

HEAT TRANSFER AND VAPOR BUBBLE DYNAMICS
IN FORCED CONVECTION BOILING

By

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NOMENCLATURE

a	vapor bubble radius (m or mm)
A	area (m^2)
C	capacitance (pf), or coulombs
C^*	relative capacitance, $(C-C_*)/(C_L-C_*)$
C_f	coefficient of friction
C_p	specific heat ($\text{J/kg}\cdot\text{K}$)
C_w	empirical constant used in Equation (6.3)
$d(t)$	vapor bubble diameter (mm or m)
d, D	width of square duct (m or mm)
d_d	vapor bubble departure diameter (m or mm)
d_L	vapor bubble lift-off diameter (m or mm)
d_w	surface/bubble contact diameter (m)
D_h	hydraulic diameter of duct (m or mm)
$f_1(r)$	probability density function for finding cavities with mouth radius r
$f_2(\psi_{mn})$	probability density function for finding cavities with minimum side angle ψ_{mn}
f	farads
F	force (N)
g	gravitational constant (9.81 m/s^2)
G	mass flux ($\text{kg/m}^2\cdot\text{s}$)
h	convection heat transfer coefficient ($\text{W/m}^2\cdot\text{K}$)

h_{fg}	latent heat of vaporization (J/kg)
H	heater thickness (mm)
Ja	Jacob number, $\rho_l C_{p,l} \Delta T_{sat} / \rho_v h_{fg}$
k	thermal conductivity (W/m·K)
K	mean vapor bubble growth constant (m/s ⁿ)
n	mean vapor bubble growth exponent
n/A	nucleation site density (cm ⁻²)
N/A	number of surface cavities per unit area (cm ⁻²)
Nu	Nusselt number
P	pressure (Pa, bar, or psi)
Pr	Prandtl number
$\dot{Q}$	volumetric flow rate (ℓ/min)
q''	heat flux (W/m ²)
q''_L	heat loss (W/m ²)
r	cavity mouth radius (mm or m)
r_{max}	maximum cavity radius allowed for nucleation (m)
r_{min}	minimum cavity radius required for nucleation (m)
R_v	universal gas constant (J/kg·K)
Re_b	bubble Reynolds number, $Re_b = 2\Delta U a / \nu_l$
Re_l	liquid Reynolds number, $Re_l = GD_b / \mu_l$
t	time (s or ms)
t_d	time of bubble departure (ms or s)
t_w	waiting time (ms or s)
T	temperature (°C or K)

$T_f(z)$	liquid temperature profile near wall ($^{\circ}\text{C}$)
u	liquid velocity (m/s)
u_t	mean liquid velocity (m/s)
$u(y)$	liquid velocity profile near wall (m/s)
u^*	friction velocity (m/s)
U	overall heat transfer coefficient ($\text{W}/\text{m}^2\text{K}$)
V	voltage (V)
V_b	bubble volume (m^3)
We_b	bubble weber number, $We_b = \rho_v \Delta U^2 2a / \sigma_t$
x	vapor quality
z	normal distance from heating surface (m)
Greek	
α	vapor volume fraction
β	receding liquid/solid contact angle
γ	microlayer wedge angle
δ	film thickness (m or mm)
ΔP	pressure difference
ΔT	temperature difference
ΔT_b	bulk wall temperature difference, $T_s - T_b$ ($^{\circ}\text{C}$)
$\Delta T_{inc, s}$	incipient wall superheat ($^{\circ}\text{C}$)
ΔT_{sat}	wall superheat, $T_s - T_{sat}$ ($^{\circ}\text{C}$)
ΔT_{sub}	bulk liquid subcooling, $T_{sat} - T_b$ ($^{\circ}\text{C}$)
ΔU	velocity difference between bubble center of mass and liquid velocity at bubble center of mass (m/s)

ϵ_r	relative permittivity ($C^2/N\ m^2 = f/m$)
θ	liquid/solid contact angle (radians)
θ_i	bubble inclination angle (radians)
λ	integral length scale (m)
λ^+	dimensionless integral time scale
μ	dynamic viscosity ($kg/m\ s$)
ν	kinematic viscosity (m^2/s)
ρ	density (kg/m^3)
σ	surface tension (N/m)
τ	integral time scale (s)
ψ_{min}	minimum cavity side angle (radians)

subscripts

1 ϕ	single-phase
2 ϕ	two-phase
a	ambient, also annular flow
b	bulk liquid
bc	bubble center of mass
i	inlet
L, ℓ	liquid
mac	macroconvection
mic	microconvection
meas	measured
pred	predicted
s	surface

sat	saturation
sub	subcooling
sup	nucleation suppression point
v	vapor
w	wall (heating surface)
x	direction parallel to heater surface
y	direction normal to heater surface

Abstract of Dissertation Presented to the Graduate School
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A forced convection boiling facility has been designed, fabricated, and tested to experimentally study small-scale heat transfer processes in vertical upflow and downflow forced convection boiling. The facility incorporates a transparent boiling test section, through which the ebullition process can be observed and recorded using digital single-frame and high speed motion cameras. A capacitance-based sensor was developed and calibrated for measurement of liquid film thickness in annular two-phase flows.

Analysis of a current model for nucleation site density has led to the development of a new dimensionless parameter, the cavity size ratio, which is demonstrated to correlate nucleation site density. Measurements of flow boiling suppression in annular flow has revealed that the cavity size ratio is, to a leading order, a fundamental parameter on which the suppression of nucleation depends. This has led to the introduction of a flow boiling regime map which delineates the purely convective and nucleate boiling regimes.

Experimental measurements of the ebullition process in single-phase inlet upflow and downflow boiling has yielded a wealth of data for vapor bubble growth rate, departure and lift-off diameters, sliding trajectories, and waiting times, in the isolated bubble regime. One very significant result of these experiments is that vastly different vapor bubble dynamics are observed for upflow, downflow, and vertical pool boiling. It is demonstrated that the differences in vapor bubble dynamics are responsible for a marked increase in the heat transfer rate for upflow compared with that of downflow.

Based on these observations, a bubble dynamic model is developed which predicts the departure diameter, sliding trajectory, and lift-off diameter in upflow, downflow, and vertical pool boiling. The model compares well to experimental data recorded in this work, as well as data reported in the literature. Furthermore, the model is applied to the case of horizontal stratified flow boiling, for which it shows a marked improvement over previous models for horizontal flow. Analysis of the model reveals, among other trends, that a negative shear lift force is responsible for maintaining vapor bubbles at the wall in upflow and ejecting bubbles from the wall in downflow.

Studies of sliding vapor bubbles have revealed that the sliding vapor bubble mechanism contributes significantly to the macroscale heat transfer in forced convection boiling. Experiments incorporating air bubble injection at the heating surface suggest that the presence of sliding bubbles enhances the bulk turbulent heat transport from the surface, and that turbulence enhancement is at least as important as the latent heat contribution to the sliding bubble heat transport mechanism.

CHAPTER 1 INTRODUCTION

Forced convection boiling, or flow boiling, is one of the most widely implemented and yet least understood modes of heat transfer. The ebullition or boiling process, subject to countless debate over the past four decades, is coupled with the turbulence structure of the bulk flow, a complex interaction which still defies fundamental understanding. As a result, engineers have historically had only simplistic models available to them to predict heat transfer rates and flow behavior. To this day engineers must rely on experience, empirical correlations, and costly prototype testing to design and analyze flow boiling systems.

The most widely used empirical method to predict flow boiling heat transfer was first suggested by Rohsenow (1952). The concept he introduced is that the rate of heat transfer associated with forced convection boiling is due to two independent and additive mechanisms: that due to bulk turbulence and that due to ebullition. Based on this concept, Chen (1966) proposed a correlation for saturated flow boiling in which the total two-phase heat transfer is obtained by simply adding the effects of macroconvection (i.e., bulk turbulence) and microconvection (i.e., ebullition). Numerous correlations based on Chen's (1966) superposition technique have been reported in the literature, many of which have been summarized by Gungor and Winterton (1986).

There are, however, serious limitations with this method. First and foremost, the concept presumes that the effects of bulk turbulence and ebullition are mutually independent. The soundness of this assumption is questionable since, in reality, the growth and motion of

vapor bubbles likely effects the hydrodynamic and thermal boundary layers, and vice versa. Second, correlations using the form proposed by Chen (1966) typically predict that the macroconvection component is the dominant heat transfer mechanism, and that the microconvection component does not contribute significantly to the total heat transfer (Klausner, 1989). In fact, numerous investigators, such as Blatt and Adt (1964), Staub and Zuber (1966), Cooper (1989), and Zeng and Klausner (1993a) have found microconvection heat transfer to be significant over a range of flow and thermal conditions. Moreover, Gungor and Winterton (1986) reported that correlations based on Chen's (1966) correlation technique fail to accurately correlate a wide range of flow boiling heat transfer data. Despite these limitations, the technique remains the most widely used to predict heat transfer rates in forced convection boiling.

The development of more reliable predictions of flow boiling heat transfer requires a thorough understanding of the fundamental mechanisms which govern the process. Forced convection boiling is comprised of a number of complicated small-scale mechanisms, interacting and combining to culminate in the macroscale heat transfer of interest to engineers. The purpose of this work is to examine these *small-scale* mechanisms to provide a framework for the development of future mechanistic models of the *macroscale* heat transfer. In the process, the superposition technique proposed by Rohsenow (1952) and used by Chen (1966) is critically examined based on the results of this study.

1.1 Small-Scale Processes in Flow Boiling Ebullition

The ebullition process in forced convection boiling heat transfer is comprised of a number of small-scale processes, which are categorized and described in the discussion which follows. Figure 1.1 illustrates an idealized vapor bubble at various stages of the ebullition

process for the case of saturated vertical upflow. For this ideal case, the behavior of the vapor bubble can be subdivided into five phases: (1) the (stationary) growth phase, in which the bubble grows while attached to the nucleation site, (2) departure (not labeled), where the bubble begins to slide away from the nucleation site, (3) sliding, where the bubble is still attached to the heater and continues to grow, (4) lift-off, where the bubble detaches completely from the surface and is ejected into the bulk flow, and (5) waiting time (not shown), defined as the time between departure and the incipience of a new bubble at the same nucleation site. These processes, along with the active nucleation site density, are necessary to characterize the heat transfer associated with the ebullition process in forced convection boiling.

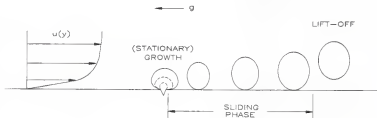


Figure 1.1. Idealized sketch of vapor ebullition process in upflow boiling.

1.1.1 Vapor Bubble Growth Rate

The rate of vapor bubble growth is a measure of the latent heat transport at the surface. To date, no adequate prediction exists for the vapor bubble growth rate in forced

convection boiling. However, for the simpler case of pool boiling, a great deal of progress has been made in the past decade toward describing the fundamental mechanisms which control bubble growth. It is therefore instructive to examine the present pool boiling concept with the aim of understanding the complexities of forced convection boiling bubble growth.

A sketch of a vapor bubble growing in an otherwise quiescent liquid pool is depicted in Figure 1.2. The vapor bubble begins as a tiny embryo in a cavity or defect in the surface of the heater. Once growth is initiated, the bubble expands very rapidly, and the momentum of the vapor is resisted on one side by the inertia of the surrounding fluid and on the other by the heating surface. The growing bubble thus deforms into a hemispherical dome which sandwiches a thin, wedge-shaped liquid microlayer between itself and the heating surface. The evaporation of the liquid within this microlayer supplies the majority of vapor to the growing bubble, and in turn removes the majority of energy from the heating surface. This rapid and localized heat transfer consumes the energy from the heater surface, resulting in temperature contours within the solid. In the case of subcooled boiling, the vapor at the top of the bubble can condense due to exposure to the cooler bulk liquid, which can ultimately lead to the collapse of the vapor bubble. In some cases evaporation and condensation may occur simultaneously, in which case the vapor bubble acts essentially as a heat pipe.

Recently, Mei et al. (1995a and b) successfully used the above concept to develop a highly detailed numerical analysis to describe the growth of vapor bubbles in saturated pool boiling. But the extension of this analysis to the case of flow boiling is complicated by several factors. First, the liquid surrounding the bubble is in motion, and is in fact turbulent motion, which depends in part on the growth and motion of the bubble. Thus both the hydrodynamic and thermal boundary layers surrounding the bubble are difficult to describe. Second, a vapor bubble exposed to forced convection is tilted by the flow, so the shape of

the bubble and the microlayer are asymmetric. Third, when sliding is present, the microlayer is continually exposed to a different part of the heater surface, and the microlayer structure itself may change in response to the bubble motion. Thus an accurate description of vapor bubble growth in flow boiling requires a detailed three-dimensional, three-phase heat and momentum transfer analysis. It is left for future investigators to attempt to model this complicated process. In this work, experimental vapor bubble growth rate data is obtained which is necessary in predicting vapor bubble detachment, and to develop a better understanding of the bubble growth process in forced convection boiling as a framework for a future analysis.

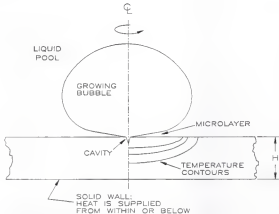


Figure 1.2. Sketch of pool boiling vapor bubble growth.

1.1.2 Vapor Bubble Detachment and Motion

The mechanisms of vapor bubble detachment in flow boiling include departure from the nucleation site, sliding, and lift-off. Following departure from the nucleation site, vapor bubbles typically slide and continue to grow until they lift off the surface. This sliding

growth continues to remove energy from the surface in two ways: First, the continued growth of the bubbles is indicative of latent heat transport. Second, the motion of the bubbles at the surface alters the local turbulence structure of the bulk liquid. Either of these mechanisms augment the heat transfer from the heating surface.

The vapor bubble sliding and lift-off mechanisms have generally been ignored in flow boiling heat transfer models reported in the literature. This is likely a result of the prevailing use of Chen-type correlations, which typically rely on pool boiling experimental data to predict the microconvection heat transfer. In pool boiling, vapor bubbles typically depart directly from the nucleation site into the quiescent liquid without sliding; consequently, the motion of the vapor bubble following departure from the site does not affect the latent heat transport at the surface. An objective of the present work is to understand and model the bubble dynamic processes, and to demonstrate to what extent the bubble sliding process affects the macroscale heat transfer.

1.1.3 Waiting Time

As stated earlier, the waiting time is defined as the time between the departure of a vapor bubble from its nucleation site and the incipience of a new bubble at the same site. It is related, along with the vapor bubble growth rate, to the energy consumption at the nucleation site, and is a partial measure of the rate of energy transport from the surface. Because the waiting time is connected with the vapor bubble growth rate, its prediction is beyond the scope of this work. Instead, an objective of this work is to experimentally examine the waiting time as a framework for future analysis.

1.1.4 Nucleation Site Density

A nucleation site is a microscopic pit or cavity on the surface of a heater in which vapor bubbles may be generated during boiling. Therefore the number of nucleation sites which actively produce vapor bubbles have a substantial influence on the ebullition heat transfer from the surface. During flow boiling, bulk turbulent convection strongly affects the nucleation site density by removing sufficient energy from the heating surface to deactivate nucleation sites. In fact, the bulk turbulent convection can be great enough to suppress nucleation completely. One objective of the present investigation is to understand the fundamental mechanisms controlling nucleation site density and suppression of sites in flow boiling.

1.2 Outline of Current Investigation

An experimental and analytical study has been performed to examine a range of small-scale processes in forced convection heat transfer and their influence on the macroscale heat transfer. An experimental facility, described in Chapter 2, has been designed, tested, and calibrated which allows visual examination of the ebullition process in upflow and downflow boiling. Both single-phase subcooled and two-phase annular flow regimes are attainable. The facility is fully instrumented to measure relevant heat transfer and flow properties, and is equipped with digital single-frame and high-speed video cameras to capture the ebullition process in a transparent boiling section. A capacitance-based sensor, described in Chapter 3, has been developed and installed to measure liquid film thickness in annular two-phase flows. The experimental facility has been calibrated and fully tested to ensure the accuracy of the measurements.

In Chapter 4, an investigation is presented of nucleation site density and the suppression of active nucleation sites in flow boiling. The nucleation site density model

proposed by Yang and Kim (1988) is examined, leading to the development of a dimensionless parameter, the cavity size ratio, which is demonstrated to correlate flow boiling nucleation site density. Furthermore, experiments conducted for annular upflow and downflow forced convection boiling suggest that this parameter is the leading order term on which the suppression of nucleation sites depends. Experiments in this portion of the investigation focused on annular flow boiling because of its relevance to a majority of industrial heat transport processes.

Chapter 5 presents an extensive experimental study of vapor bubble growth and motion in single-phase inlet vertical upflow and downflow boiling. Using high-speed digital imaging, a series of experiments were performed to measure vapor bubble growth rates, departure diameters, sliding trajectories, lift-off diameters, and waiting times in subcooled single-phase inlet upflow and downflow under various liquid mass fluxes and wall superheats. The condition chosen for this portion of the investigation were restricted to single-phase flow inlet to the test section, and focused primarily on the isolated bubble boiling regime. These conditions were selected in order to facilitate high-speed photography, as well as to eliminate the effects of adjacent sites on the dynamics of the isolated bubbles. Furthermore, the absence of annular flow eliminates interfacial effects, which simplifies the modeling of bubble behavior.

A thorough examination of the data is also presented in Chapter 5, providing a wealth of insight into the ebullition process and its effect on the macroscale heat transfer in vertical flow boiling. Photographs of ebullition reveal vastly different vapor bubble dynamic behavior under upflow, downflow, and vertical pool boiling conditions, which helps to explain the difference in heat transfer rates observed between upflow and downflow boiling. Examination of the vapor bubble growth rate, departure and lift-off diameters, and waiting

time demonstrate the complicated interaction of the bulk flow and the energy transport at the heater surface. From these data the mechanisms which control the growth and motion of the vapor bubble are postulated, and a framework is developed of a phenomenological vapor bubble dynamics model.

In Chapter 6, the knowledge obtained from experiments is used to develop a vapor bubble dynamics model for vertical upflow and downflow. The model predicts vapor bubble departure and lift-off diameters, as well as sliding trajectories, utilizing a force balance similar to that of Zeng et al. (1993a and b) which was developed for horizontal flow boiling. The model compares well to experimental data from a range of flow and thermal conditions, including data taken from the literature. Moreover, the model is demonstrated to predict features of bubble lift-off behavior in horizontal flow which were not captured by Zeng's (1993a and b) model.

In Chapter 7, a detailed investigation of the vapor bubble sliding process is presented which quantifies the contribution of vapor bubble sliding energy transport to the total heat transfer in forced convection boiling. Experiments performed in upflow and downflow over a range of flow conditions reveal that the vapor bubble sliding mechanism contributes significantly to the macroscale heat transfer. Experiments using air bubble injection are performed to isolate the effect of bulk turbulence enhancement on the bubble sliding energy transport mechanism. The results of these tests demonstrate that bulk turbulence enhancement may be as important as the latent heat transport in the bubble sliding energy transport mechanism.

The work concludes in Chapter 8 with a summary of this work, as well as a discussion of the critical phenomenological issues which remain unresolved. Finally, recommendations are proposed for future research efforts.

CHAPTER 2 EXPERIMENTAL FACILITY

2.1 System Overview

The experimental work in this research was performed in a vertical flow boiling test facility designed and fabricated under the direction of the author. A schematic diagram of the system is shown in Figure 2.1. A variable speed Model 221 Micropump drives the working fluid through the facility. Following the pump, the fluid is filtered to eliminate contaminants. The volumetric flow rate is then measured using either a venturi- or vane-type flow meter, each installed to measure a different range of flow rates. The fluid is preheated via four preheaters to achieve a saturated two-phase mixture, then routed by a series of valves (1a&b, 2a&b) to obtain vertical upflow or downflow in the test section. The square test section, described in detail in Section 2.2, features a DC-powered nichrome heating strip, and is used to measure wall temperatures and heat fluxes, as well as to make visual observations of nucleation and hydrodynamic phenomena using a CCD camera. A capacitance sensor, described in Chapter 3, is used to measure film thickness in annular two-phase flows. From there the flow enters a water-cooled, shell and tube condenser/receiver which returns the fluid to a subcooled state.

The operation of the facility is achieved through a combination of manual control and automatic monitoring. The physical operation of the system, i.e. the pump, preheaters, test section, and cooling via the condenser/receiver, is controlled manually. Nearly all measurements of temperature, pressure, flow rate, etc., are made using a data acquisition system controlled by a PC-style computer. An elaborate computer code was written which outputs both unprocessed

data and calculated quantities on the screen in real time, so that complete monitoring of the system data is possible during the experiment.

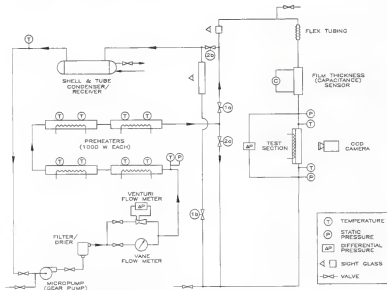


Figure 2.1. Schematic diagram of vertical flow boiling facility.

FC-87, a perfluorocarbon fluid supplied by 3M Corporation, was used as the working fluid for the system. This fluid was selected for several reasons: First, its low latent heat of vaporization (26.9 kJ/kg @ 25°C) reduces the heat input required to achieve nucleate boiling. Second, its low boiling point (30.0 °C @ 1 atm) translates to a lower system temperature and pressure, allowing more flexibility in the selection of working materials and the design of components. Additionally, the fluid is nontoxic, chemically inert, and poses no threat to the ozone layer. Finally, the properties of FC-87 are well documented; a list of thermodynamic and physical properties is provided in Appendix A.

2.2 Development of Test Section

2.2.1 Test Section Design

The most critical component of the boiling facility is the transparent test section, where nucleate boiling phenomena is measured and observed. Several requirements govern the design; first, the test section is used with the optical imaging system to conduct high-resolution measurements, so the section must be transparent and provide a clear and unimpeded view of the heating/boiling surface and flow. Second, a method is needed to attach the nichrome heating strip to an inside wall of the section, and to attach leads through the walls to an external power source. Thermocouples must also be attached to the underside of the heater, while being electrically insulated from the heater, and penetrate the walls of the section. Finally, the test section must withstand the operating pressures and temperatures of the system, and attach to the rest of the system without disturbing the flow. All of these requirements must be met while maintaining leak-free operation.

The design of the test section underwent several iterations before success in all aspects was achieved; the final design is depicted in Figure 2.2. The walls of the test section and the heater assembly are constructed of 1/2-inch-thick cast Lexan plate; these are bonded together with methacrylate resin to form a 12.7×12.7 mm square duct. Cast Lexan was chosen for its optical clarity and its strength. A 30 cm long, 0.15 mm thick nichrome strip, clamped and adhered to one wall, is used as a heating surface. A detailed discussion of the heater assembly is presented in Section 2.2.2. Two 1-inch-thick Lexan flanges are sleeved over the ends of the duct and bonded with the resin. The test section was leak-tested to 414 kPa (60 psig); leaks were sealed by lowering the pressure inside the test section with a vacuum pump, and using suction to fill the leaks with 5-Minute epoxy. The test section is installed to the boiling facility as shown in Figure 2.3. The section is connected at both ends to 12.7×12.7 mm ID square brass tubing, extending approximately 1 meter above and below the test section to form a hydrodynamic development

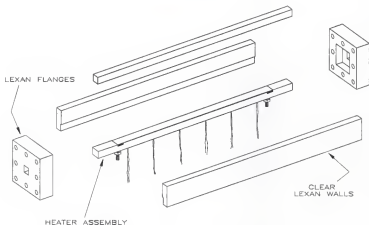


Figure 2.2. Exploded view of visual test section.

length for the flow. A brass plate, soldered to the end of the square tube, is machined to form a double O-ring face seal. This type of flange seals the joint while leaving the flow undisturbed. A series of flange seals are also used to install the capacitance sensor, and to connect the square tubing with the rest of the facility. In addition, a flexible tube is mounted to the top of the line to aid in alignment, and to eliminate stresses on the test section and capacitance sensor due to thermal expansion.

Power to the heater is supplied by a 240 volt/20 Amp/3-phase AC source, converted to 36 V/125 A DC through an autotransformer and rectifier. A rheostat and autotransformer are manually adjusted to control the DC power input to the heater. Two digital voltmeters measure the power into the heater: the voltage across the heater is measured directly, while the current is determined by measuring the voltage across a 1 m Ω shunt connected in series with the heater.

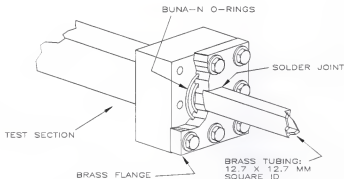


Figure 2.3. Connection of test section to system.

2.2.2 Fabrication of Heater Assembly

The fabrication of the heater assembly is shown in Figure 2.4. The ends of the heater are wrapped over two brass bolts, machined with flat square heads. The ends are then sandwiched between the bolts and square brass washers, which clamp the heater in place when attached to the Lexan base. Two hex nuts hold the clamps in place, while O-ring washers seal the protrusions. The brass bolts, protruding outside, attach to an external DC power source. To measure the heater surface temperature, six equally-spaced thermocouples are attached to the underside of the nichrome strip, and lead out through holes in the base. A detailed description of the thermocouples used in the facility are described in Section 2.4.6.

The most difficult aspect of the design is the attachment of the thermocouples to the heater and the attachment of the heater to the Lexan base. In prior designs, numerous epoxies, resins, and glues were used to bond the heater to the base, but thermal expansion would cause the nichrome strip to shear off of the adhesive and detach completely from the Lexan base. Ultimately the technique illustrated in Figure 2.5 proved successful. A high-strength, non-

hardening adhesive tape manufactured by 3M Corporation holds the heater to the base; it forms a flexible bond which allows thermal expansion of the heater. The heater and base are slightly wider than the duct, and the joining walls are machined with a 0.5 mm lip which holds the edges of the heater in place. The heater is also pretensioned while adhered to the base to counteract thermal expansion. The thermocouples are attached to the heater with Hysol Aerospace Epoxy, at spots pre-coated with the epoxy to provide electrical insulation. The stripped thermocouple wire is sealed within the base with 5-Minute epoxy. A small air space near the thermocouple junction is provided to eliminate shear stress due to thermal expansion of the heater.

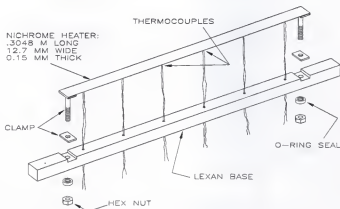


Figure 2.4. Exploded view of test section heater assembly.

2.3 Air Injection System

An air injection system was installed at the entrance to the test section to allow for the injection of air bubbles at the heater surface during certain experiments. A schematic diagram of this system is depicted in Figure 2.6. The nozzle is constructed of 1.6 mm OD brass tubing, crimped at the tip to provide a small (less than 100 μm) opening. The tubing is soldered to the square brass channel containing the working fluid, such that the nozzle tip extends into the boiling

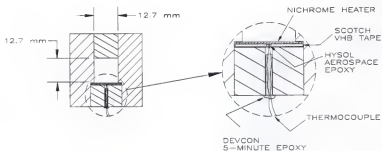


Figure 2.5. Cross-section of test section. Detail of heater attachment.

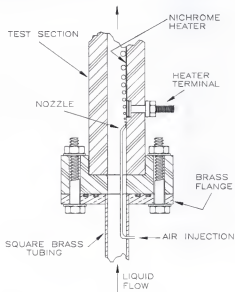


Figure 2.6. Detail of transparent test section with air injection nozzle.

test section just below the heater. Compressed air is supplied to the nozzle, which is adjusted by a needle valve to obtain air flow rates ranging from zero to 50 ml/min. A capillary-tube flow meter was constructed in the laboratory to measure air flow rate, where the pressure drop across the 0.83 mm ID capillary tube is measured by a Validyne model DP45-16 differential pressure transducer. The voltage output of the transducer was calibrated using a syringe pump, and the estimated uncertainty of the air flow rate measurement is ± 2.0 l/min. Prior to entering the test section, the air is first routed through approximately 2 meters of 3.2 mm OD copper tubing, attached to the square brass tubing attached to the inlet of the test section. This technique allows the air to be preheated by the inlet liquid flow to ensure that the air enters the test section at the bulk fluid temperature.

2.4 Instrumentation and Calibration

2.4.1 Flow Meters

Two flow meters are installed in the facility to measure a wide range of volumetric flow rates. A venturi meter is used to measure low flow rates (0.0-1.8 l/min). The differential pressure from the entrance to the throat is measured with a Validyne model DP15 variable reluctance differential pressure transducer. The electrical signal of the transducer is converted to a voltage output using a carrier demodulator, which is then connected to a data acquisition system discussed in Section 2.5. The voltage is calibrated directly against the volumetric flow rate, using a volume-time method. The resulting calibration curve is depicted in Figure 2.7. A curve fit of the form $y = Ax^*$ yields

$$\dot{Q} = 0.54537 V^{0.52017} \quad (2.1)$$

where $\dot{Q}$ is the volumetric flow rate in liters per minute, and V is the voltage across the transducer. To account for a small drift in the zero-flow-rate reading, the voltage is referenced

to a zero reading taken at the beginning of each test. The standard deviation of the experimental data for the above curve fit is 0.43% of full scale, within the accuracy of the DP15 as reported by the manufacturer.

An Erdco model 2521 vane-type flow meter is installed parallel to the venturi meter to obtain a higher range of flow rates (3.0-9.0 ℓ/min). Two valves installed prior to the meters allow the user to switch between the two. The vane meter is equipped with a 4-20 ma analog output, which is connected to a 500 ohm power resistor. The voltage across the resistor is recorded by the data acquisition system, and calibrated against the volumetric flow rate as before. The resulting calibration curve for this meter is depicted in Figure 2.8. A third-order polynomial curve fit of the experimental data yields

$$\dot{Q} = 1.067 + 1.156V - 0.02354 V^2 + 0.004055 V^3 \quad (2.2)$$

where $\dot{Q}$ is in liters per minute and V is in volts. The standard deviation of the experimental data for this curve fit is 0.44%, which is within the limits of repeatability reported by the manufacturer.

2.4.2 Preheater Heat Loss

Four preheaters raise the fluid from a subcooled liquid state to the desired vapor quality prior to entering the test section. Each heater, one of which is illustrated in Figure 2.9, is coiled around a 3/8 inch copper tube, and delivers up to 1000 Watts. Each is powered by a 240 AC line, controlled manually through an adjustable AC autotransformer. The power input to each of the preheaters is measured and recorded manually: a digital volt meter measures the voltage across each heater, while a clamp-on AC current probe is used with a digital multimeter to measure the current. The system is insulated, from the preheaters to the entrance to the condenser/receiver, using 25 mm thick fiberglass pipe insulation. Heat loss through the insulation is considered negligible except at the preheaters, where it must be determined. To

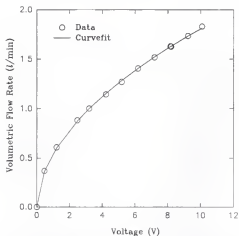


Figure 2.7. Calibration curve for venturi flow meter.

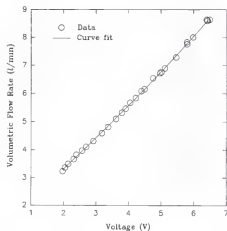


Figure 2.8. Calibration curve for vane flow meter.

calibrate this heat loss, the working fluid is drained from the system, and a known power is input to the preheaters. When thermal equilibrium has been reached so that the power input equals the heat loss through the insulation, thermocouples record the surface temperature of the insulation, T_s , along with the ambient temperature, T_a . When this is repeated over various power inputs, a curve of heat loss versus $T_s - T_a$ is generated, which is depicted in Figure 2.10. A fourth-order polynomial curve fit of the data gives

$$q_L = -0.07275 + 0.6618\Delta T + 0.07057\Delta T^2 - 0.002863\Delta T^3 + 0.00004101\Delta T^4 \quad (2.3)$$

where q_L is in Watts and $\Delta T = T_s - T_a$, with T in degrees Celsius.

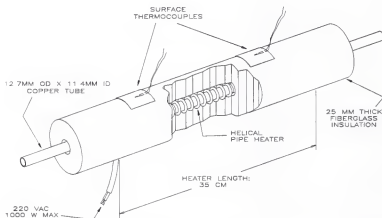


Figure 2.9. Preheater assembly.

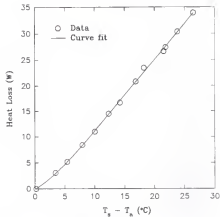


Figure 2.10. Preheater heat loss calibration curve.

2.4.3 Test Section Heat Loss

In order to accurately measure the energy imparted to the flow in the test section, the energy loss from the test section to the atmosphere must be taken into account. To quantify this effect, the following procedure was used. At a flow rate of approximately $250 \text{ kg/m}^2\text{-s}$ and a given system operating temperature, the facility was allowed to come to thermal equilibrium, and measurements of bulk flow temperature and heater surface temperatures were recorded. Assuming a 2-dimensional control volume depicted in Figure 2.11(a), an energy balance at the fluid/wall interface yields

$$q_1'' = q_2'' \quad , \quad (2.4)$$

or

$$h(T_s - T_f) = U(T_s - T_a) \quad (2.5)$$

where h is the convective heat transfer coefficient of the fluid, U is the overall heat transfer coefficient for the heat transfer from the fluid/wall interface to the atmosphere, and T_b , T_s , and T_a are the bulk flow, surface, and ambient temperatures, respectively. Equation (2.5) implies that, if the temperatures are measured and the convective heat transfer coefficient for the fluid flow can be determined, the overall heat transfer coefficient, U , may be readily determined.

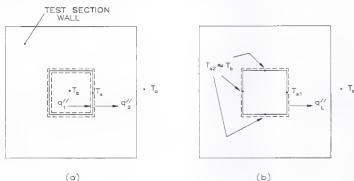


Figure 2.11. Control volumes for calibration of heat loss from test section.

A plot of $(T_b - T_a)$ vs. $(T_s - T_a)$ over a range of bulk fluid temperatures is illustrated in Figure 2.12. Linear regression curve-fits of each set of data are included in the figure. The $(T_b - T_a)$ data were measured relative to both thermocouples which measure bulk flow temperature, one located directly above and one below the test section. The slope of both curve-fits are approximately equal, although a small difference in the magnitude of the two curves ($\pm 0.4^\circ\text{C}$) is noted. This is attributed to the difference in bulk temperature, T_b , measured by the two thermocouples; this difference is within the stated accuracy of the thermocouples. The slope of either curve-fit is thus a measure of the ratio of U/h . To estimate h for single-phase flow, the model by Petukhov (1970) is used for turbulent, fully developed flow through a duct,

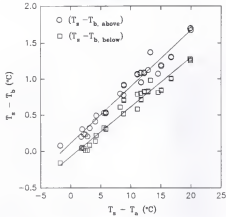


Figure 2.12. Calibration of temperature difference between bulk flow and heater surface.

$$Nu_{D_h} = \frac{hD_h}{k} = \frac{Re Pr(C_f/2)}{1.07 + 12.7(Pr^{2/3} - 1)\sqrt{C_f/2}} \quad , \quad 10^4 < Re < 5 \times 10^6 \quad (2.6)$$

$$\frac{C_f}{2} = (2.236 \ln Re - 4.639)^{-2} \quad , \quad 0.5 < Pr < 2000 \quad (2.7)$$

where D_h is the hydraulic diameter of the duct which is equal to the side width for a square cross-section, C_f is the coefficient of friction, Re is the Reynolds number, and Pr is the Prandtl number of the fluid. Estimating the properties of FC-87 to be constant, the heat transfer coefficient for the fluid flow for these experimental data is estimated to be $h \approx 300 \text{ W/m}^2\text{K}$. Finally, from the slope of the curves in Figure 2.12, the overall heat transfer coefficient across the test section to the atmosphere is calculated from Equation (2.5) to be $U = 22.0 \text{ W/m}^2\text{K}$.

During operation, an energy balance on the test section gives

$$\dot{q}'' - \dot{q}_L'' = h(T_s - T_b) \quad , \quad (2.8)$$

where q'' is the power input to the heater per unit surface area of the heater, $q_L'' = U(T_s - T_\infty)$ is the heat loss, and $h(T_s - T_b)$ is the energy convected from the heater by the fluid flow. the convection coefficient is not typically known a priori and must be determined using Equation (2.8).

When power is supplied to the heater, the inner surface temperature of the test section is no longer uniform, since only one wall of the test section is heated. Therefore in order to estimate the heat loss, an average surface temperature is calculated by assuming the unheated surfaces to be approximately equal to the bulk temperature. This condition is depicted in Figure 2.11(b). A weighted average of surface temperatures is then used to calculate q_L'' in Equation (2.8).

2.4.4 Test Section Pressure Drop

The streamwise pressure drop for the test section is measured with a Validyne model DP15 variable reluctance differential pressure transducer, identical to the one used with the venturi flow meter. A carrier demodulator converts the signal output of the device to a voltage, which is recorded by the data acquisition system. The voltage signal is proportional to the differential pressure; the calibration curve, depicted in Figure 2.13, results in the linear expression

$$\Delta P = 1.377V \quad (2.9)$$

where ΔP is given in kPa and V in volts. As before, some drift is encountered with the voltage output at zero ΔP , so the voltage measured is referenced to a zero- ΔP voltage taken at the beginning of each test. The standard deviation of the calibration data for the above curve fit is 0.05% of full scale, which is within the accuracy stated by the manufacturer.

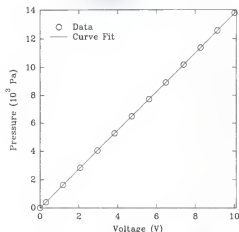


Figure 2.13. Calibration curve for test section differential pressure.

2.4.5 Static Pressure Measurements

Two Viatran model 2415 static pressure transducers are installed at either side of the test section to measure inlet and outlet pressure and, in conjunction with thermocouples mounted nearby, to calculate thermophysical properties of the working fluid. Transducers number 1 and 2, mounted below and above the test section, respectively, are calibrated in Figures 2.14 and 2.15. The resulting linear curve fits are

$$P_1 = -1.551 + 41.63 V \quad (2.11)$$

below the test section, and

$$P_2 = -5.502 + 69.10 V \quad (2.12)$$

above the test section, where P_1 and P_2 are given as gage pressure in kPa and V is in volts. Both calibrations are accurate to 0.12% of their respective full scale.

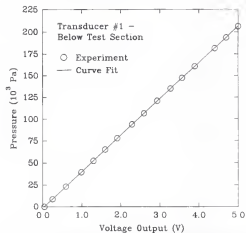


Figure 2.14. Calibration curve for static pressure transducer #1.

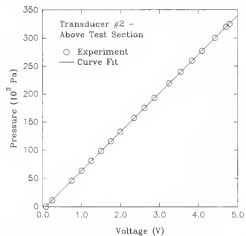


Figure 2.15. Calibration curve for static pressure transducer #2.

2.4.6 Temperature Measurements

Numerous temperatures are monitored and/or recorded during an experimental run. The temperature at the inlet to the preheaters establishes the subcooled liquid state. Surface temperatures on the preheater insulation determine the heat loss at the preheaters. Temperature probes at either inlet (depending on the flow direction) of the test section establish the bulk fluid two-phase temperature, and are used with the temperature at the preheater inlet to evaluate the vapor quality of the flow. Finally, temperatures on the surface of the test section heater are monitored.

Temperature measurements throughout the system were made with Type E (Chromel-Constantan) thermocouples, accurate to within $\pm 1.0^{\circ}\text{C}$. 36-gauge wire (0.005 in./0.127 mm dia.) was selected for fast response. Thermocouple junctions were welded in the laboratory, resulting in a typical bead size of approximately 0.44 mm. Thermocouple probes, manufactured in the laboratory, were used to measure bulk fluid temperature. The probes were constructed of 1/16 inch brass tubing; the thermocouple wire was inserted into the tubing such that the junction was exposed directly to the flow. The wire was sealed inside the tubing using epoxy adhesive, and the probe was installed in the system with brass compression fittings. The voltage output of the thermocouples was measured by the data acquisition system and converted to temperature via operating software (described in Section 2.5).

2.5 Data Acquisition System

A digital data acquisition system has been developed for monitoring and recording measurements of pressure, temperature, flow rate, and capacitance during the experiment. The system is comprised of two major components: the data acquisition hardware and the operating software. An ACCES AD12-8 12-bit, 8-channel analog-to-digital (A/D) converter board installed in an 80386 personal computer performed the data acquisition. Two ACCES AIM-16 16-channel multiplexer cards are interfaced to two channels on the board, allowing the system to measure

up to 32 different signals. Each channel of the multiplexers has a preamplifier with programmable gains ranging from 0.5 to 1000. A thermistor located at channel 0 of the multiplexers served as the reference temperature for thermocouple measurements. The A/D board and the multiplexer cards were calibrated according to the manufacturer's specifications.

A QuickBASIC computer code, developed for this work, operates the data acquisition hardware and gives continuous data output of every instrument during an experiment. The operation of the system is as follows: each analog signal from the respective instrument was supplied to one of the 32 channels of the multiplexers. Appropriate gains were set in the program for each channel to achieve maximum signal resolution. Because two-phase flows are inherently unstable, and because signals received from thermocouples and other devices carry unwanted noise, all measurements were time-averaged to obtain repeatable values. Typically, 500 sampling points were taken for each channel over a 5-second interval and averaged to obtain one set of data.

The operating software converts the voltage data of the sensors to pressure, temperature, flow rate, and capacitance using standard correlations and those developed in Section 2.4. A menu-driven screen format was developed to display the data. The software also displays film thickness, vapor volume fraction, vapor quality, and other calculated values for monitoring during the experiment. A keystroke changes the format of the output from converted data to unprocessed voltages to allow verification of instrument operation. Finally, all unprocessed data, reduced data, and calculated properties and quantities are saved at the operator's command to ASCII file format, which is imported to spreadsheet software for further analysis and graphical display.

2.6 Imaging Facilities

Two digital imaging facilities were used to obtain images of the flow and of nucleation phenomena through the transparent boiling test section. The first consists of a Videk Megaplus

CCD single-image camera with image resolution up to 1320×1036 pixels. The camera is equipped with a Vivitar 50 mm macro lens for high magnification and low optical distortion. The output of the CCD camera is input to an Epix 4 megabyte frame grabber card installed in an 80386 personal computer. The frame grabber allows for either high resolution (1320×1035 pixels) or low resolution (640×480 pixels) images, displayed on a Sony monitor with 1000 lines/inch resolution. The imaging acquisition was software-controlled using a C language algorithm, and image processing and analysis were performed using two commercial programs, Pixfolio and SigmaScan.

A Kodak EktaPro digital high-speed motion camera was used to capture moving images of the ebullition process. The Model VSG Intensified Imager and Controller captures images with a resolution of 192×239 pixels at rates ranging from 1000 full-size frames per second up to 6000 partial frames per second. Typically, half-frame images were recorded at 2000 frames per second. The camera is equipped with a Vivitar 50 mm macro lens for high magnification and low distortion. The lens/camera configuration results in an image field of approximately 2×4 mm (half-frame images) with a $20 \mu\text{m}$ resolution. A typical synchronous recording is comprised of 1600 images spanning 800 ms in real time. An EktaPro EM Motion Analyzer stores the images, which are then downloaded to standard S-VHS video tape. A description of the image acquisition and analysis procedure is described further in later chapters.

CHAPTER 3 DEVELOPMENT OF CAPACITANCE-BASED FILM THICKNESS SENSOR

3.1 Introduction and Literature Survey

Capacitance sensors have recently gained popularity for measuring vapor volume fraction in two-phase flows. Their popularity is primarily due to the fact that they are non-invasive and are simple to operate. However, the design of capacitance sensors can vary greatly depending on the desired temporal and spatial resolution, signal-to-noise ratio, two-phase flow pattern, and temperature variations. For this reason, prior investigators have put forth great effort in (i) understanding those factors that dictate the performance of capacitance sensors, and (ii) improving the performance of those sensors. Most prior work has been conducted for two-phase flows through circular tubes, but many of the results can be generalized to other geometries. The focus of previous work may be categorized as described below.

Sensor geometry. The sensitivity of the capacitance sensor to changes in liquid film thickness or vapor volume fraction strongly depends on the geometry of the sensing electrodes, and various investigators have examined the performance of different electrode configurations. Irons and Chang (1983) and Chun and Sung (1986) tested strip and ring type electrodes for gas/powder and liquid/vapor flows, respectively. Gregory and Mattar (1973) and Abouelwafa and Kendall (1980) tested many different electrode configurations for liquid/vapor flows, including helical, strips, parallel plates, and concave plates. Ozgu and Chen (1973) considered ring-type and parallel-strip type electrodes and discussed the required spacing between electrodes to achieve the optimum sensitivity. These studies revealed, among other things, that the calibration of a capacitance sensor is flow regime dependant. Therefore the anticipated flow

regime, whether known or unknown, should be considered in choosing the electrode configuration.

Analytical/numerical modeling. Analytical attempts have been made to estimate the sensor output for various flows. Abouelwafa and Kendall (1979a, 1979b) estimated the output of helical and concave-plate type sensors for homogeneous, slug, annular, and stratified flows by assuming a uniform charge distribution on diametrically opposed electrodes as well as uniform flux lines between them. Chun and Sung (1986) used a similar approach for ring-type electrodes. Geraets and Borst (1988) solved for the three-dimensional potential field between helical electrodes and developed a closed form expression for the capacitance associated with annular and dispersed flows. Gupta et al. (1994) used a finite-difference technique to solve for the potential field between concave plates with stratified flow. Although they used this solution to compute the impedance, it may also be used to compute the capacitance.

Spatial sensitivity. Xie et al. (1990) conducted a detailed investigation on the effect of electrode size, wall thickness, and the proximity of electrical shielding on the spatial sensitivity of a concave plate capacitance sensor. Based on their analysis, they attempted to optimize the electrode design so that the sensor output is approximately linear and independent of flow regime. However, due to non-uniformities in the electric field with changes in flow regime, it is not possible to design a sensor with a purely linear output. Klug and Mayinger (1994) used an eight-electrode sensor to measure impedance in horizontal gas/liquid flows. The use of capacitance was also considered to measure two-phase flows of nonconducting fluids. The impedance was measured between various pairs of electrodes, each sensitive to a different region of the flow. The impedances measured from these pairs is compared to those produced by known flow patterns in order to identify flow regime and volume fraction.

Temporal resolution for dynamic measurements. Various authors have used the time history of the capacitance signal output as a way of discerning information about the flow

structure. Geraets and Borst (1988) used the signal time history as a way of characterizing the flow pattern in horizontal gas-liquid flows. Klausner et al. (1992) examined the power spectral density function for the capacitance signal to study the low-frequency fluctuations of film thickness in horizontal stratified and annular liquid-vapor flows. Keska et al. (1994) also studied frequency and power spectral density distributions, from time traces of concentration, film thickness, and pressure, in horizontal air-water flow.

Effect of temperature variations. Klausner et al. (1992) discussed the effect of temperature variations on both the permittivity of the phases and the surrounding sensor body and devised a temperature-correction scheme to account for the effect.

In this work, a capacitance-based film thickness sensor is designed, and an experimental investigation and analysis are performed to predict its output as a function of liquid film thickness. Both stratified and annular flow patterns are considered. Since the fluids considered have a low relative permittivity (FC-87, $\epsilon_r = 1.72$; FC-72, $\epsilon_r = 1.75$), the analysis was very useful in designing a sensor with satisfactory resolution. Furthermore, the analysis eliminates the need for sensor calibration when measuring horizontal stratified flow or annular flow film thickness. The analysis is validated by measurements of stratified, stationary films in the sensor. The sensor is then installed in the vertical boiling facility, where annular flow film thickness measurements are obtained with the digital imaging facility and compared against those measured using the capacitance sensor. The temperature range over which the flow boiling facility operates is 20 to 80°C, and thus it is necessary to account for non-isothermal conditions, which is accomplished by calibrating the sensor over a range of operating temperatures.

3.2 Sensor Design and Instrumentation

An exploded view of the square cross section capacitance sensor is illustrated in Figure 3.1. The sensor is constructed in a similar manner to the test section, with four 12.7 mm thick Lexan walls, chemically welded with methacrylate resin. Two Lexan flanges welded at each end

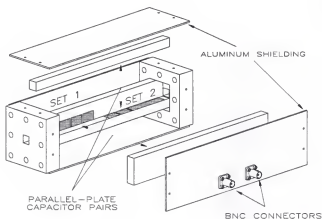


Figure 3.1. Exploded view of capacitance meter (wiring omitted for clarity).

of the sensor allow for connection to the flow boiling facility. Two sets of parallel brass plates, mounted apart and rotated 90° from each other, span the width of the channel and are attached to the inside surfaces of the walls. The result is two separate sensors, labeled as 'set 1' and 'set 2' in Figure 3.1, which are mounted in orthogonal planes. The resulting capacitances of the respective sets are electrically connected in parallel such that the signals are combined. This technique helps to ensure that asymmetries in the film thickness, if present, do not affect the combined signal from the sensor. In each set, the plates comprising the capacitor are divided into 3 separate pieces, one long plate (12.7×60 mm) and two short plates (12.7×20 mm). The plates can be wired together in different combinations, allowing different effective electrode lengths, and therefore allowing some control over the sensitivity of the sensor. In this study, however, only the long electrodes are used.

Surrounding the sensor are 4 aluminum plates, mounted to the sides of the flanges, each grounded to shield the electrodes from external electromagnetic fields. The two sets of capacitors

are connected together (in parallel for the current configuration) with shielded cable, then connected to BNC fittings mounted on the aluminum plate. The wiring of the sensor has been omitted in the figure for clarity. The sensor is connected via BNC fittings and shielded cable to a Keithley 590 digital CV analyzer. The instrument measures capacitance with a resolution of 0.1 fF and accurate to 0.1% of full scale. An analog signal from the analyzer is input to the data acquisition system for monitoring and recording. In order to test the adequacy of the capacitance sensor analysis, the CCD camera is used to photograph the liquid film through a visual section. The liquid film thicknesses measured using the capacitance sensor are compared with CCD camera film thickness measurements.

3.3 Methodology of Sensor Analysis

An analysis is performed on two parallel plates of area A , separated by a distance d , with a homogeneous material of relative permittivity ϵ_r in between. For the ideal case where $\sqrt{A} \gg d$, the electric field is constant throughout the material, and the capacitance is given by

$$C = \frac{\epsilon_0 \epsilon_r A}{d} \quad , \quad (3.1)$$

where $\epsilon_0 = 8.85 \times 10^{-12} \text{ C}^2/(\text{N} \cdot \text{m}^2) = 8.85 \text{ pF/m}$ is the permittivity of free space. Due to the linearity of the Laplace equation which governs the potential field within the sensor, the capacitance of a composite material in the space between the plates can be analyzed by constructing an equivalent circuit of series and parallel capacitances. For example, the composite material shown in Figure 3.2 has a capacitance

$$C = C_1 + \left[\frac{1}{C_2} + \frac{1}{C_3} + \frac{1}{C_2} \right]^{-1} + C_1 \quad (3.2)$$

where C_1 , C_2 , and C_3 are computed using Equation (3.1) applied to the equivalent geometry shown.

The composite structure depicted in Figure 3.2 is equivalent to a steady two-phase annular flow where the vapor core has a square cross section. Of course, in a real flow, the annular core is never completely square in cross section. When the vapor volume fraction is small the annular core approaches a circular shape, although for small vapor volume fractions annular flow is not likely to occur. And as the volume fraction approaches unity the core shape approaches a square with rounded corners. Given this reasoning, the calibration procedure assumes that the annular core cross section will be square.

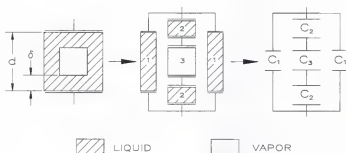


Figure 3.2. Reduction of square vapor core flow geometry into composites.

Nevertheless, it is of interest to determine the degree of error introduced in the vapor volume fraction measurement when the shape of the core is circular. To address this concern, the method of composites, described above, is repeated for the case of a round vapor core geometry. The capacitance sensor is divided into very small composites, as demonstrated in Figure 3.3. For the same vapor volume fraction, the computed capacitance for the round vapor core may be compared against that of a square core. The percent difference between the computed capacitance for the two geometries is depicted in Figure 3.4. The comparison is made

up to a vapor volume fraction of 0.785, which corresponds to the largest circular vapor core that can exist in a square cross section. It is seen that, when the vapor volume fraction is small, the error in approximating the round vapor core as square becomes vanishingly small. As the vapor volume fraction increases, the error increases. However, two points are worth noting. First, as the vapor volume fraction increases, the shape of the core will begin to conform to the shape of the square channel. Indeed, for annular flow conditions achievable in the current two-phase flow facility ($\alpha > .80$), the vapor core appears to be a square with rounded corners. Second, the maximum error in Figure 3.4 is less than 0.30%, which is within the accuracy of the measuring technique. It is noted, however, that the small error is partially due to the fact that the ratio of the liquid to vapor permittivity is small, $\epsilon_{liq}/\epsilon_{vap} = 1.75$. Therefore for this work, the vapor core geometry can be effectively approximated as square without the introduction of any significant error.

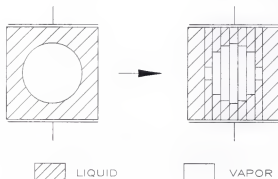


Figure 3.3. Reduction of round vapor core capacitance into composites.

3.4 Results and Discussion

3.4.1 Stratified Film

It is desirable to test the capacitance film thickness sensor on a bench top where the liquid film can be precisely controlled and measured. For liquid/vapor mixtures it is not possible to do so for the annular flow regime. So instead, the sensor was tested on the bench top using FC-87 ($\epsilon_r = 1.72$) for a stratified liquid layer. These tests validate the composite material analysis, which will subsequently be used to evaluate the liquid film thickness for annular flow. The usefulness of this methodology for evaluating the annular flow film thickness will be demonstrated in the following section.

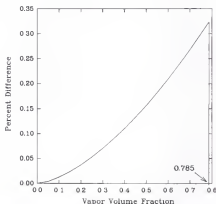


Figure 3.4. Percent difference in capacitance between annular flow with a square and round vapor core.

The square channel may be placed in two different orientations: the upwards orientation and the side orientation, as shown in Figure 3.5. C_u and C_s denote the capacitances measured for a stratified liquid layer in the respective upward and side orientations. Assuming that (3.1) is valid, the ratio of the liquid to vapor permittivity can be expressed as

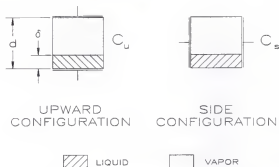


Figure 3.5. Upward and side configurations of parallel-plate capacitor in stratified flow.

$$\frac{\epsilon_L}{\epsilon_v} = \frac{C_L}{C_v} \quad (3.3)$$

where C_L is the measured capacitance when the sensor is filled with pure liquid and C_v is the measured capacitance when the sensor is filled with pure vapor, and the ratio of C_L to C_v is referred to here as the effective permittivity ratio. Using Equations (3.1) and (3.3) and the composite material analysis, the dimensionless capacitances for a stratified liquid layer in the upward and side orientations, as shown in Figure 3.5, may be expressed as

$$C_u^* = \frac{C_u - C_v}{C_L - C_v} = \frac{1}{\frac{C_L}{C_v} - 1} \left(\frac{\frac{C_L}{C_v}}{\frac{\delta}{d} + \frac{C_L}{C_v} \left(1 - \frac{\delta}{d} \right)} - 1 \right) \quad (3.4)$$

and

$$C_s^* = \frac{C_s - C_v}{C_L - C_v} = \frac{\delta}{d} \quad (3.5)$$

where δ is the liquid film thickness and d is the channel width.

Figures 3.6 and 3.7 show the predicted film thickness as a function of capacitance for the upward and side orientations respectively. Also shown are the measured film thicknesses using the CCD camera associated with the measured capacitances using the parallel plate sensor. The capacitances were measured for only one set of parallel plates. The estimated uncertainty of the CCD camera measurements is $\pm .01$ mm. It is seen that there is good agreement between the analysis and the experimental measurements. The standard deviations for the measured and predicted film thickness for the upward and side orientations are 0.014 and 0.019 mm, respectively, thus validating the composite material analysis. C_u and C_s may be computed using Equations (3.4) and (3.5) for a given film thickness at any system temperature provided C_L and C_v are known as a function of temperature.

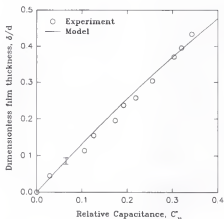


Figure 3.6. Comparison between predicted and measured film thickness for the upward orientation. Capacitance is measured using the sensor, and film thickness is measured by the CCD camera.

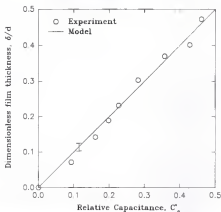


Figure 3.7. Comparison between predicted and measured film thickness for the side configuration. Capacitance is measured using the sensor, and film thickness is measured using the CCD camera.

3.4.2 Annular Flow

For the sensor design considered herein, using one set of parallel plates and FC-72 as the working fluid, the calibration curves for C_L and C_v as a function of temperature are shown in Figure 3.8. Over the temperature range considered, 23-60°C, C_L and C_v are approximately linear functions of temperature. The objective in this work is to measure the liquid film thickness for annular two-phase vertical upflow or downflow using the capacitance sensor without having to calibrate it for annular flow conditions. Utilizing the composite material analysis shown in Figure 3.2 in conjunction with Equation (3.2), the capacitance, C_a^* , for annular flow with a square cross section and uniform film thickness may be expressed as

$$C_a^* = \frac{C_a - C_v}{C_L - C_v} = \frac{\left(2 \frac{\delta}{d}\right)}{\left(\frac{C_L}{C_v} - 1\right)} \left[\frac{C_L}{C_v} - \frac{1}{2 \left(\frac{\delta}{d}\right) + \frac{C_L}{C_v} \left(1 - 2 \left(\frac{\delta}{d}\right)\right)} \right] \quad (3.6)$$

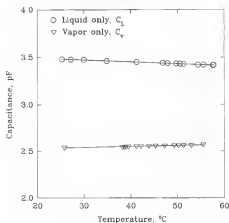


Figure 3.8. Temperature calibration of C_L and C_v .

Thus, Equation (3.6) and the calibration curves for C_L and C_v are used to evaluate the liquid film thickness, δ , for a measured annular flow capacitance, C_a .

The capacitance sensor was installed above the test section in the vertical flow boiling facility charged with FC-72. For a given system pressure, two-phase flow was established by evaporating liquid FC-72 in a preheater section. There are approximately 120 diameters of adiabatic developing length for the liquid/vapor mixture prior to entering the capacitance sensor. The flow boiling facility is configured such that either vertical upflow or downflow may be established in the capacitance sensor. Both sets of electrodes were used for the annular flow tests. Directly adjacent to the capacitance sensor is a visual test section, through which the liquid film can be observed. The system mass flux and vapor quality are adjusted to obtain annular two-phase flow. The liquid film in two-phase flow is inherently unstable and thus the film displays large temporal and spacial variations. Typical CCD photographs of the annular flow liquid film for vertical upflow and downflow are shown in Figures 3.9(a) and (b), respectively.

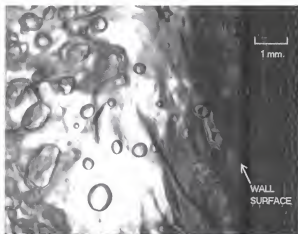


Figure 3.9(a). Typical photograph of liquid film in annular upflow.

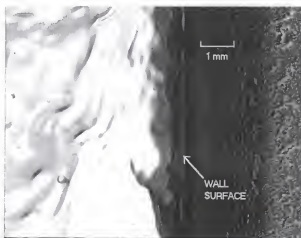


Figure 3.9(b). Typical photograph of liquid film in annular downflow.

An ensemble of ten photographs were used to evaluate the liquid film thickness in vertical downflow using the CCD camera; twenty photographs were taken for upflow due to the highly variable nature of the film. A comparison between the film thickness measured from photographs and a time-averaged measurement using the capacitance sensor is shown in Figure 3.10. The estimated uncertainty of the photographic film thickness measurements is $\pm .014$ mm for upflow and $\pm .019$ mm for downflow. Considering the large temporal and spatial variations in film thickness observed with two-phase flow, the comparison is satisfactory. It is seen that the largest deviation is for vertical upflow. This is because the liquid film for upflow is considerably more variable than that for downflow, and obtaining an accurate measurement of the film thickness from photographs is quite difficult. It is believed that the average film thickness measurements using the capacitance sensor are more reliable than those using the camera. The standard deviation for the downflow comparison is 0.093 mm and that for upflow is 0.173 mm.

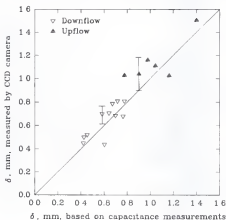


Figure 3.10. Comparison of annular two-phase flow liquid film thickness measured with capacitance sensor and CCD camera.

CHAPTER 4

NUCLEATION SITE DENSITY AND SUPPRESSION

4.1 Introduction and Literature Survey

Over the past four decades, the heat transfer rates associated with forced convection boiling have been investigated by hundreds of researchers. Perhaps the first investigators to propose a correlation for predicting flow boiling heat transfer rates were Dengler and Addoms (1956). Since then many other flow boiling heat transfer correlations have been proposed, many of which have been summarized by Gungor and Winterton (1986). The type of correlation which has been used the most was first proposed by Chen (1966) and takes the form

$$h_{2\phi} = h_{mac} + h_{mic} \quad (4.1)$$

where $h_{2\phi}$ is the two-phase heat transfer coefficient, h_{mac} is the macroconvection heat transfer coefficient due to bulk turbulent transport, and h_{mic} is the microconvection heat transfer coefficient due to the ebullition process. Klausner (1989) examined various correlations which take the same form as (4.1) and found they predict that the macroconvection component of heat transfer is the dominant heat transfer mechanism, and the microconvection component of heat transfer typically does not contribute significantly to the total two-phase heat transfer. One notable exception is the correlation proposed by Kandlikar (1990), which predicts that microconvection can be a considerable portion of the total heat transfer. Zeng and Klausner (1993a) measured the microconvection component of heat transfer for horizontal flow boiling of refrigerant R113 and demonstrated that under certain flow and thermal conditions it contributes significantly to the total two-phase heat transfer, as demonstrated in Figure 4.1. Others who have reported finding

microconvection heat transfer to be significant over a range of flow and thermal conditions include Blatt and Adt (1964), Staub and Zuber (1966), Frost and Kippenhan (1967), Kenning and Cooper (1989), Cooper (1989), Klausner (1989), and Tran et al. (1993).

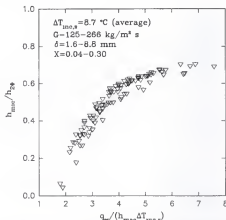


Figure 4.1. Ratio of microconvection heat transfer coefficient to total two-phase heat transfer coefficient for different flow boiling conditions (from Zeng and Klausner; 1993a).

In order to understand the discrepancy, it is instructive to consider the saturated vertical upflow boiling experiments by Kenning and Cooper (1989). Over the range of flow and thermal conditions tested, two flow boiling regimes were identified: the apparently convective regime and the apparently nucleate boiling regime. In the apparently convective regime, the measured two-phase heat transfer coefficient appears to be independent of the heat flux and heating surface conditions and can be predicted within an uncertainty of approximately $\pm 10\%$. In this regime, it appears that the boiling activity has been completely suppressed. In the apparently nucleate boiling regime, the measured heat transfer coefficients are essentially independent of mass flux, vapor quality, and flow regime. They tend not to be repeatable due to variations in surface conditions and gas entrainment, and they are difficult to predict.

The importance of the two flow boiling regimes identified by Kenning and Cooper cannot be underestimated. When nucleation sites are completely suppressed and heat transfer is dominated by bulk turbulence and liquid film evaporation, as occurs in the convective regime, the physics governing the heat transfer are completely different than that in the nucleate boiling regime in which heat transfer is mainly controlled by the incipience, growth, and departure of vapor bubbles. As demonstrated in Figure 4.1, as soon as the wall superheat is sufficient such that incipience occurs, microconvection strongly contributes to the total two-phase heat transfer. Moreover, without a clear distinction between the two regimes, experimental data used to construct empirical heat transfer correlations can contain convective data interspersed with nucleate boiling data. This can bias a correlation toward one regime, making it incapable of satisfactorily predicting the heat transfer in the other. Therefore it is of critical importance to be able to predict which heat transfer regime is present for a given set of flow conditions.

The objective of this work is to understand the fundamental mechanisms controlling the suppression of nucleation sites in flow boiling inside channels or tubes with net vapor generation, which establishes the transition point between the convective and nucleate boiling regimes. Based on thermodynamic relations and physical reasoning, a dimensionless parameter is identified to serve as a criterion for predicting when nucleation sites become completely suppressed. It is tested and calibrated using experimental heat transfer data obtained in this work, flow boiling nucleation site density data of Zeng and Klausner (1993b), and heat transfer data of Kenning and Cooper (1989) and Jallouk (1974). The working fluids considered are water, R113, R114, and FC-87.

4.2 Nucleation Sites in Flow Boiling

An extensive experimental investigation of nucleation site density in horizontal flow boiling with refrigerant R113 was reported by Zeng and Klausner (1993b). They demonstrated that the behavior of the nucleation site density is significantly different than that found in pool boiling,

and although the nucleation site density data show dependence on the critical (minimum) cavity radius, it is insufficient for correlating the data. They observed a strong dependence on heat flux and mean vapor velocity. As the mean vapor velocity increased, the nucleation site density decreased until complete suppression of nucleation sites occurred. No generalized correlation for the data was suggested. In general the formation of nucleation sites is highly dependent on the size and geometry of the microscopic scratches and pits on the heating surface, the wettability of the fluid, the amount of foreign contaminants on the surface, and the material from which the surface was fabricated. Here, the data of Zeng and Klausner (1993b) are reexamined based on theoretical considerations.

It has long been recognized that surface cavities which become active nucleation sites must be capable of trapping vapor (Corty and Foust, 1955). Wang and Dhir (1993) suggested that only those cavities with $\theta > \psi_{min}$ are capable of trapping vapor, in which θ is the liquid-solid contact angle and ψ_{min} is the minimum cavity side angle, or in the case of spherical cavities, ψ_{min} is the mouth angle. Using this vapor trapping criterion and the nucleation site density model proposed by Yang and Kim (1988) for pool boiling, the nucleation site density for either pool or flow boiling may be calculated from,

$$\frac{n}{A} = \frac{N}{A} \int_{r_{min}}^{r_{max}} f_1(r) dr \int_0^{\theta} f_2(\psi_{min}) d\psi_{min} \quad (4.2)$$

where n/A is the nucleation site density, N/A is the number of cavities per unit area on the heating surface, f_1 is the probability density function of finding cavities with mouth radius r , f_2 is the probability density function of finding cavities with minimum side angle ψ_{min} , and r_{min} and r_{max} are the respective minimum and maximum mouth cavity radii within which incipience will occur. The use of Equation (4.2) assumes a statistically homogeneous distribution of cavities on the heating surface and statistical independence of r and ψ_{min} .

Although the applicability of (4.2) to flow boiling has yet to be experimentally verified, it is quite useful in understanding the nucleation site density data presented by Zeng and Klausner (1993b), in which n/A is observed to decrease rapidly with increasing vapor velocity. Increasing the mean vapor velocity results in enhanced two-phase bulk turbulence and thus reduces the thermal boundary layer thickness. The temperature field which is seen by a vapor embryo protruding a surface cavity, as sketched in Figure 4.2, is strongly influenced by the bulk turbulence. A vapor embryo will grow and establish a nucleation site only if there is sufficient energy from the surroundings to sustain growth. If a vapor embryo is too large, which might occur in a cavity with a large diameter, it will protrude into the colder region of the liquid, which reduces the vapor temperature to below the value required by the Clapeyron equation for the equilibrium of the two phases. Therefore, in addition to a minimum cavity radius required for incipience, there also exists a maximum cavity radius, above which the incipience is inhibited. As the bulk turbulence is enhanced, the thermal boundary layer becomes thinner and $r_{max} \rightarrow r_{min}$, and thus the nucleation sites become suppressed as suggested by (4.2). The distinct difference in the formation of nucleation sites between pool and flow boiling lies in the fact that r_{max} in flow boiling is typically much less than that in pool boiling due to a much thinner thermal boundary layer. In fact, Wang and Dhir (1993) were able to adequately predict pool boiling nucleation site density by assuming $r_{max} \rightarrow \infty$. As will be demonstrated, such an assumption is not valid for flow boiling.

4.2.1 Estimating the Minimum and Maximum Cavity Radii

In order to estimate r_{min} and r_{max} for saturated flow boiling, consideration is given to a hemispherical vapor embryo which protrudes a surface cavity as shown in Figure 4.2, for which the following assumptions are made. (1) The static vapor embryo is embedded in a saturated liquid shear flow with a temperature profile governed by macroconvection heat transfer. (2) The embryo radius is typically much less than the liquid film thickness, and the temperature profile

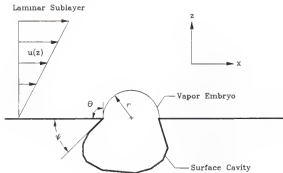


Figure 4.2. Idealized sketch of a vapor embryo embedded in shear flow protruding a surface cavity.

in the vicinity of the embryo may be approximated as linear. Following Zeng and Klausner (1993a), the near-wall temperature profile, $T_t(z)$, is approximated as

$$T_t(z) = T_w - \frac{h_{mic} z}{k_t} \Delta T_{sat} \quad (4.3)$$

where T_w is the mean wall temperature, ΔT_{sat} is the wall superheat, and k_t is the thermal conductivity of the liquid. During vapor bubble growth under saturated conditions, energy is transferred to the vapor through the liquid microlayer which resides beneath the bubble. However, prior to growth no liquid microlayer can be established and thus energy is transferred to the embryo from the surrounding liquid and the solid heating surface beneath the embryo. A physically sound criterion for vapor embryo growth to continue is that there must exist net heat transfer into the embryo from the surroundings. Resolution of the temperature field in the vicinity of the embryo requires a three-dimensional, three-phase heat transfer analysis. Because of the complexity of such an analysis, the problem is greatly simplified by approximating the liquid temperature profile near the wall to be linear as given by (4.3), and the incipience criterion

proposed by Bergles and Rohsenow (1964) is instituted: vapor bubble growth will proceed only when the liquid temperature, T_l , at the top of the hemispherical embryo exceeds the vapor temperature, T_v . It is assumed that the vapor embryo temperature T_v is uniform, which is computed from the Clapeyron equation as

$$T_v - T_{sat} = \frac{T_v T_{sat} R_v}{h_{fg}} \ln \left(1 + \frac{2\sigma}{r P_{sat}} \right) \quad (4.4)$$

where R_v is the engineering gas constant, σ is the surface tension, P_{sat} is the pressure in the liquid phase, and h_{fg} is the latent heat. Thus the maximum and minimum cavity radii which can sustain an active nucleation site may be estimated by setting $T_l = T_v$ and $z = r$ in (4.3) and solving (4.3) and (4.4) for r . An approximation for r can be made by noting that $\ln \left(1 + \frac{2\sigma}{r P_{sat}} \right) = \frac{2\sigma}{r P_{sat}}$ for many cases of practical interest. Implementing the perfect gas law and combining (4.3) and (4.4) results in

$$\frac{h_{mac} r}{k_\ell} = 1 - \frac{T_{sat} 2\sigma}{h_{fg} \rho_v \Delta T_{sat} r} \quad (4.5)$$

Equation (4.5) is quadratic in r , and the two solutions give the respective maximum and minimum cavity radii, r_{max} and r_{min} . Simplification of the solutions to (4.5) may be made by noting that $(T_{sat} 2\sigma h_{mac} / h_{fg} \rho_v \Delta T_{sat} k_\ell)$ is small compared with unity for many cases of interest, and thus approximate expressions for r_{max} and r_{min} are:

$$r_{max} = \frac{k_\ell}{h_{mac}} \quad (4.6)$$

and

$$r_{min} = \frac{2\sigma T_{sat}}{\Delta T_{sat} h_{fg} \rho_v} \quad (4.7)$$

A straightforward result of this analysis is the identification of the dimensionless parameter r_{max}/r_{min} , which is referred to as the cavity size ratio,

$$\frac{r_{\max}}{r_{\min}} = \frac{\Delta T_{\text{sat}} k_t \rho_v h_{fg}}{h_{\text{mac}} T_{\text{sat}} 2 \sigma} \quad (4.8)$$

In cases where the approximations leading to Equations (4.6)-(4.8) cannot be satisfied, it is only necessary to use an iterative technique to simultaneously solve Equations (4.3) and (4.4) for $r_{\max}$ and $r_{\min}$. It is noted that variations of the Bergles-Rohsenow incipience criterion have appeared in the literature, but inherent assumptions are used in all of these. In light of the assumptions used in the incipience criterion, the assumptions used to arrive at Equation (4.8), and the nonuniform geometry of most nucleation sites, $r_{\max}/r_{\min}$ should be viewed as a dimensionless variable on which the formation and suppression of nucleation sites depends.

As implied by (4.2), the probability of finding cavities on the heating surface which have mouth radii which fall between $r_{\max}$ and $r_{\min}$ decreases as $r_{\max} \rightarrow r_{\min}$. Specifically, it is noted that $r_{\max}/r_{\min} = 4$ is the limit allowed by the Bergles-Rohsenow incipience criterion. This suggests that $r_{\max}/r_{\min}$ may serve as a useful correlating parameter for the nucleation site density. In fact, when $r_{\max}/r_{\min}=4$ it is not physically possible to maintain heterogeneous boiling; however, this is too restrictive a condition to use as a suppression criterion since it is possible for nucleation sites to become completely suppressed for $r_{\max}/r_{\min} > 4$, depending on the distribution of cavity mouth radii on the heating surface. It is expected that the topography of the heating surface plays some role, as do the wetting characteristics of the fluid. The usefulness of $r_{\max}/r_{\min}$ as a dimensionless variable to characterize nucleation suppression will be demonstrated empirically.

4.2.2 Nucleation Site Density

Figure 4.3 shows the flow boiling nucleation site density data, measured by Zeng and Klausner (1993b) using R113 in horizontal stratified flow over a smooth nichrome heating strip, displayed as a function of $r_{\max}/r_{\min}$ for two different heat fluxes, $q_w = 19.3$ and 13.8 kW/m^2 . In these data the mass flux, vapor quality, liquid film thickness, and wall superheat varied. The lines through the data are merely curve fits. In computing $r_{\max}/r_{\min}$, experimental values for $b_{\max}$

were used. It is seen that for a given heat flux the data collapse into a single curve. It is noted that Zeng and Klausner were unable to correlate their nucleation site density data with r_{min} alone. For the data shown in Figure 4.3, r_{min} does not change appreciably, while r_{max} varies significantly. Thus the variation in r_{max}/r_{min} is primarily controlled by r_{max} , which suggests that r_{max} has a controlling influence on flow boiling nucleation site density. This is significant because r_{max} has historically been ignored in correlating n/A .

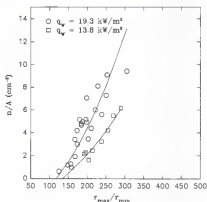


Figure 4.3. Nucleation site density data of Zeng and Klausner (1993b) shown as a function of r_{max}/r_{min} .

According to (4.2), the nucleation site density as a function of r_{max}/r_{min} should not be heat flux dependent. The heat flux dependence shown in Figure 4.3 may be due to the fact that (4.3) and (4.8) assume a uniform wall temperature. In fact, in the presence of heterogeneous boiling, the thermal field on the heating surface is highly nonuniform with large temperature gradients beneath nucleation sites, which are more pronounced at higher heat fluxes. As the nucleation site density is reduced, the temperature nonuniformities should be less pronounced and the observed convergence of the data in Figure 4.3 is expected. The heat flux dependence displayed in Figure 4.3 may also be a result of the simplified incipience criterion to used compute r_{max}/r_{min} .

When nucleation is completely suppressed, the wall temperature should be uniform (within the length scale of the bubble size) and the two curves fitted to the data are expected to intersect at $n/A=0$. Although they do not exactly intersect at $n/A=0$, $(r_{max}/r_{min})_{sup}=117$ for $q_w=19.3$ kW/m², which is close to $(r_{max}/r_{min})_{sup}=130$ for $q_w=13.8$ kW/m², where the subscript sup denotes the point of suppressed nucleation.

A result which makes clear the limitations of the Bergles-Rohsenow incipience criterion is that $(r_{max}/r_{min})_{sup}$ is much greater than unity. Although the magnitude of r_{max}/r_{min} is high, its value as a correlating parameter will be further demonstrated. In order to further test the applicability of correlating the cavity size ratio with the point of suppressed nucleation, a vertical forced convection boiling facility has been constructed in which measurements of the flow boiling nucleation suppression point have been made for vertical upflow and downflow over a wide range of flow and thermal conditions. A perfluorocarbon fluid, FC-87, having similar surface wetting characteristics as R113, was used as the working fluid, and nichrome was again used as the heating surface. Suppression point data have been obtained for mass flux, G , ranging from 183 to 315 kg/m²-s; inlet quality, x_i , ranging from 0 to 0.151; and heat flux, q_w , ranging from 1.20 to 16.95 kW/m².

4.3 Experimental Procedure

The nucleation site density suppression point data were obtained from the boiling facility described in Chapter 2 in the following manner: for a given mass flux and inlet vapor quality at the test section, sufficient power was applied to the heater to achieve vigorous boiling. The power input to the test section heater was then stepped down in small increments, and measurements of heat flux, mass flux, wall superheat, vapor quality, liquid film thickness, and saturation pressure and temperature were obtained. Decreasing wall heat flux was used in order to avoid boiling hysteresis. A flow boiling curve (q_w vs. ΔT_{sat}) was generated for the given flow conditions. Typical flow boiling curves for vertical upflow and downflow are illustrated in

Figures 4.4(a) and (b), respectively. The suppression point is determined from the flow boiling curve by observing that in the convective regime the heat transfer coefficient is independent of heat flux, and q_w varies linearly with ΔT_{sat} . The suppression point is therefore taken to be the point at which the heat flux curve departs from linearity as shown in Figures 4.4(a) and (b). The value of ΔT_{sat} for which nucleation suppression is established has an estimated uncertainty of $\pm 0.5^\circ\text{C}$. The macroconvection heat transfer coefficient, h_{mc} , is the slope of the q_w vs. ΔT_{sat} curve in the convective regime. The macroconvection heat transfer coefficient, h_{mc} , is determined with an uncertainty of $\pm 5\%$. In order to validate the heat transfer coefficient measurements, single-phase flow heat transfer coefficients were measured over a range of Reynolds number. The results are compared against Petukhov's (1970) single-phase flow heat transfer correlation in Figure 4.5. It is seen that the agreement is good.

4.4 Results

In the present work, a set of 29 suppression points were determined from vertical upflow and downflow boiling curves. Using flow conditions and properties obtained from the experiment, the cavity size ratio at the point of suppression, $(r_{max}/r_{min})_{sup}$, was calculated using (4.8). In determining $(r_{max}/r_{min})_{sup}$, experimental values of h_{mc} (the slope of the q_w vs. ΔT_{sat} curve in the convective regime) were used. A summary of the suppression point data and thermal and flow conditions at the point of suppression are displayed in Table 4.1.

Figures 4.6 to 4.8 show the variation of $(r_{max}/r_{min})_{sup}$ for vertical upflow and downflow with mass flux, inlet quality, and wall heat flux, respectively. According to (4.2), $(r_{max}/r_{min})_{sup}$ should be independent of these parameters. Furthermore, for a specific heating surface and fluid combination, $(r_{max}/r_{min})_{sup}$ should be invariant. As shown in Figures 4.6 to 4.8, the measured values for $(r_{max}/r_{min})_{sup}$ are scattered slightly over a narrow range, $40 < (r_{max}/r_{min})_{sup} < 60$, and show no dependence on inlet quality and very little dependence on mass flux, which qualitatively supports the theory. However, Figure 4.8 shows a slight dependence of $(r_{max}/r_{min})_{sup}$ on heat flux,

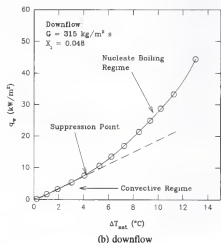
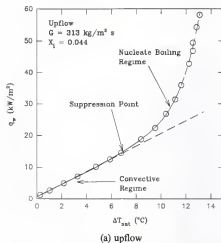


Figure 4.4. Boiling curves for vertical annular upflow and downflow, illustrating nucleation suppression point.

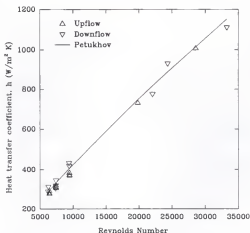


Table 4.1. Point of suppressed nucleation heat transfer data from present work.

G (kg/m ² -s)	x_i	δ (mm)	q_w (kW/m ²)	ΔT_{sat} (°C)	h_{mac} (W/m ² °C)	$(t_{\text{max}}/t_{\text{min}})_{\text{exp}}$
<i>upflow</i>						
200	0.045	1.60	16.18	8.60	2022	54.8
201	0.052	1.68	16.36	8.01	2043	53.2
200	0.057	1.54	16.62	8.25	2015	56.6
201	0.087	1.41	15.52	7.54	2059	50.5
201	0.120	1.21	16.23	7.75	2094	51.5
200	0.151	1.11	15.55	7.46	2084	49.5
260	0.023	1.75	16.21	7.94	2041	54.2
261	0.062	1.39	16.51	7.66	2155	50.0
260	0.087	1.31	16.27	7.58	2147	50.2
261	0.109	1.15	16.67	7.49	2225	48.7
262	0.137	1.07	13.67	8.17	2285	52.5
311	0.025	1.65	15.57	7.41	2101	50.1
314	0.044	1.48	16.36	7.58	2158	50.6
314	0.067	1.31	16.50	7.48	2206	49.5
314	0.102	1.14	17.52	7.41	2364	47.3
<i>downflow</i>						
183	0.060	0.851	7.00	5.12	1367	51.8
202	0.062	0.829	8.15	5.07	1603	44.5
236	0.054	0.892	8.52	4.97	1714	41.2
237	0.075	0.889	6.53	4.21	1550	38.5
238	0.128	0.860	6.04	4.05	1492	39.9
237	0.151	0.829	6.82	4.43	1540	43.0
284	0.023	0.940	8.06	4.48	1798	36.0
280	0.048	0.939	7.91	4.56	1734	38.3
280	0.076	0.927	9.19	5.05	1819	39.6
280	0.109	0.900	8.23	4.82	1707	42.6
315	0.026	0.981	9.41	4.74	1986	34.6
314	0.049	0.968	8.78	4.67	1881	36.6
313	0.073	0.951	8.19	4.50	1819	37.0
311	0.106	0.919	7.73	4.30	1798	36.8

Table 4.2. Point of suppressed nucleation from Kenning and Cooper (1989) heat transfer data.

$(r_{max}/r_{min})_{sup}$	q_w (kW/m ²)	G (kg/m ² -s)
108	50	304
90	100	304
103	100	123
111	150	123
74	200	304
74	200	203
99	250	123
69	300	304
89	300	123
63	350	123

It is seen that the data demonstrate a slight dependence on heat flux, but are generally scattered between $(r_{max}/r_{min})_{sup}=40$ -120. In general, the point of suppressed nucleation will depend on the fluid/surface combination and the wetting characteristics of the fluid; however, despite vastly different flow configurations and fluid/surface combinations, Equation (4.8) yields suppression point cavity size ratios within the same order of magnitude. In this study, both highly wetting fluids (FC-87, R113, R114) and an intermediate wetting fluid (water) have been considered. As indicated in Figure 4.9, three regimes may be identified based on r_{max}/r_{min} ,

Regime I: $r_{max}/r_{min} < 40$, Convective Regime (suppressed nucleation)

Regime II: $40 \leq r_{max}/r_{min} \leq 120$, Regime of Uncertainty (nucleation may or may not be sustainable)

Regime III: $r_{max}/r_{min} > 120$, Nucleate Boiling Regime (nucleation is sustained).

Thus Figure 4.9 may be used as a flow boiling regime map to determine whether the flow and thermal conditions correspond to sustained nucleation or purely forced convection. The purpose of the flow boiling regime map is to provide guidance to practitioners, experimenters, and theoreticians as to which regime is most probable for specific operating conditions.

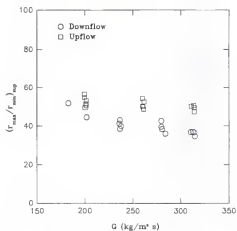


Figure 4.6. Suppression point cavity ratio, $(r_{\max}/r_{\min})_{\text{sup}}$, as a function of mass flux.

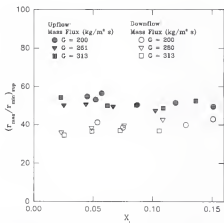


Figure 4.7. Suppression point cavity ratio, $(r_{\max}/r_{\min})_{\text{sup}}$, as a function of test section inlet quality.

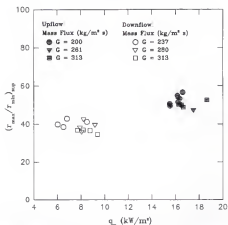


Figure 4.8. Suppression point cavity ratio, $(r_{max}/r_{min})_{sup}$, as a function of wall heat flux.

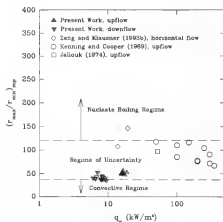


Figure 4.9. Identification of two-phase heat transfer regimes based on point of suppressed nucleation.

4.5 Discussion

The cavity size ratio, r_{max}/r_{min} , has been demonstrated to correlate nucleation site density as well as distinguish between the convective and nucleate boiling regimes in flow boiling. Experimental data from this work, along with data obtained from the literature, are used to examine $(r_{max}/r_{min})_{exp}$ and identify the transition between the two regimes. The values of $(r_{max}/r_{min})_{exp}$ are much greater than unity; however, data taken over a wide range of flow conditions, heating surfaces, and working fluids all fall within the same order of magnitude. This suggests that the cavity size ratio is, to the leading order, the most important dimensionless parameter in characterizing the suppression of nucleation sites.

The cavity size ratio identified in this work is intended as a first step toward a better understanding of flow boiling nucleation and the development of a flow boiling regime map. This work also provides direction for future flow boiling nucleation investigations. An incipience criterion based on conservation of energy which provides a realistic description of the thermal boundary layer in the vicinity of the vapor embryo may give an improved estimate for r_{max} . Equation (4.2) dictates that the heating surface topography and fluid/surface wetting characteristics play a role in the formation and suppression of nucleation sites, both of which require improved understanding.

CHAPTER 5 OBSERVATION AND MEASUREMENT OF VAPOR BUBBLE DYNAMICS

5.1 Introduction and Literature Survey

The high heat transfer rates associated with forced convection boiling are due to two complex processes working together: bulk turbulent convection and the growth and detachment of vapor bubbles at the heating surface. Therefore in order to develop reliable predictive capabilities for forced convection boiling heat transfer, a fundamental understanding of the ebullition process in the presence of a turbulent flow is essential. However, while numerous measurements of vapor bubble growth rate and departure diameter have been reported in the literature for nucleate pool boiling, the available flow boiling data are limited. For this reason among others, pool boiling heat transfer correlations are often incorporated into those for flow boiling.

In recent years, great strides have been made toward understanding vapor bubble dynamics in horizontal flow boiling. Klausner et al. (1993) and Zeng et al. (1993) developed a vapor bubble departure and lift-off model for horizontal flow boiling, which agrees well with a wide range of data obtained using high-speed imaging. However, a general extension of their model to vertical flow boiling remains uncertain. For the vertical downflow configuration, sliding bubbles lag the liquid velocity, and the resulting shear lift could be sufficient to lift vapor bubbles off the heating surface. However, quantifying the shear lift force requires knowledge of the bubble sliding velocity, which is not provided by the model. For vertical upflow, the sliding bubble velocity leads the liquid velocity, and thus the shear

lift pushes the bubble against the heating surface. Whether a mechanism exists to remove vapor bubbles from the heating surface remains uncertain. In the case of microgravity flow boiling, a mechanism for removing vapor bubbles from the heating surface has yet to be identified, and thus it is not clear under what conditions microgravity flow boiling can be sustained.

Several investigations have sought to visually document the ebullition process in forced convection boiling. Gunther (1951) used high-speed cinematography to examine vapor bubble dynamics in subcooled, upflow boiling of water. Vapor bubble radii were measured during the growth and collapse stages in subcooled flow, and the effects of flow velocity and subcooling were documented. Gunther observed that the vapor bubbles slide away from the nucleation site, and remain attached to the heating surface throughout the growth and collapse stages. Hsu and Graham (1963) performed a similar visual study with highly subcooled water to investigate the hydrodynamic features of vertical bubbly and slug flow regimes subjected to boiling. No measurements of bubble growth rate or departure diameter were made. However, bubbles ejected into the bulk flow, and "ejection" diameters were measured. The authors attributed the ejection, in part, to a lifting force imposed by the flow. Frost and Kippenhan (1967) examined the growth and collapse of bubbles in subcooled upflow of water containing various concentrations of a surface active agent. Bubble growth and departure diameters were measured. Cooper et al. (1983) measured bubble growth and displacement in supersaturated *n*-hexane in forced laminar upflow over a stationary wall. Similar measurements were made for short duration microgravity flow. Recently, van Helden et al. (1995) obtained growth and departure diameter data in saturated and superheated upflow boiling of pure water. He reported sliding to occur following departure, but did not record the bubble trajectory.

The purpose of this work is to experimentally examine the complex vapor growth and removal process in vertical upflow and downflow forced convection boiling, with an emphasis on understanding the role of gravity. An experimental facility which uses a dielectric fluid, FC-87, has been designed, tested, and calibrated and allows visual examination of the ebullition process under slightly ($1.0\text{--}5.0^{\circ}\text{C}$) subcooled conditions. Using a high-speed digital video imaging facility, a visual study of the bubble dynamics in the isolated bubble regime is presented. Image analysis of the film reveals vapor bubble growth, departure diameter, sliding trajectory, lift-off diameter, and waiting time. In Chapter 6, the detailed forces acting on a vapor bubble are considered, and a vapor bubble departure and lift-off model is proposed which is an extension of that developed by Zeng et al. (1993). Included in the model is a prediction of the bubble sliding trajectory, which is also compared to experimental data.

5.2 Experimental Procedure

The experimental facility described in Chapter 2 was operated to achieve single-phase flow through the transparent boiling test section. The bulk single-phase conditions at the entrance of the test section are controlled by adjusting the preheat and flow rate. Once steady-state flow is established, power is supplied to the test section heater at a sufficient rate to obtain vigorous boiling at the surface. The heat flux is then reduced until only isolated nucleation sites are active. Operating the facility in this manner eliminates the effect of boiling hysteresis.

The Ektapro high-speed digital camera is mounted on a tripod, and is aimed at the heater from the side at approximately a $25\text{--}30^{\circ}$ angle above the heater. The test section was backlit (the lighting was aimed at the camera) through a gray filter to create a diffusely illuminated flow. In order to achieve the highest possible resolution and eliminate errors in

calibration, the camera lens is fixed at a constant focal length, resulting in a fixed viewing area of approximately 4×4 mm (full-frame). To view an image in the test section, the camera is moved forward or backward using a linear traverse installed on the head of the tripod. This technique eliminates the need to recalibrate the image magnification every time the field of view is adjusted. Image analysis is performed manually by measuring distances on the film as it is played back from video tape on a 20-inch video monitor with 1000 lines per inch resolution. The lateral displacement of the vapor bubble is the distance measured from the nucleation site to the bubble centroid. The characteristic bubble diameter is taken to be the length of the chord measured through the centroid of the bubble and parallel to the wall. This definition is chosen because the camera angle normal to the heater must be varied slightly in order to achieve an unobstructed view of a nucleation site, which introduces uncertainty as to the image magnification in the normal direction. Moreover, thermal gradients in the boundary layer and reflection of the bubble image on the heater surface make the location of the base of the bubble difficult to distinguish precisely. However, as will be shown, the bubbles are very nearly spherical during the early portion of growth and sliding, during which the diameter is measured. Therefore the bubble diameter as it is measured is judged to be an adequate representation of the bubble size.

An important consideration in these experiments involves the temporal and spatial variations in the heater surface temperature and liquid velocity. Specifically, it is necessary to estimate the time scale of these variations to ensure that the film length is sufficient to establish statistically meaningful values of mean bubble growth, departure and lift-off diameter, and sliding trajectory. For surface temperature, variations due to ebullition can be estimated using the time taken for growth and departure plus the waiting time. For experimental conditions examined in this work, this time scale is typically on the order of

20 to 50 ms. Velocity fluctuations may be estimated as follows. From a numerical simulation of single-phase turbulent channel flow (Lyons, 1989), the integral length scale in the streamwise direction is on the order of $\lambda^+ = \lambda u^*/\nu_t \sim 500$. Thus an estimate of the time scale, assuming the frozen field hypothesis is valid, is on the order of $\tau = (500\nu_t/u^*)/u_t \sim 10^{-1}$ s, where u_t is the mean liquid velocity in the duct. Considering both of these estimated time scales, the 800 ms film length is judged to be sufficient to capture the statistical variations in the measured bubble growth, departure and lift-off diameter, and trajectory. To ensure consistency of the film data, two to three films are acquired at each condition.

5.3 Flow Visualization

Table 5.1 summarizes the experimental conditions and data collected for vertical upflow and downflow forced convection boiling considered in this work. A total of 20 experimental conditions were studied, with mass fluxes ranging from 193 to 666 kg/m²-s and heat fluxes ranging from 1.3 to 14.6 kW/m². The bulk flow was slightly subcooled, with ΔT_{sub} ranging from 1.9 to 5.2°C. In addition, one vertical surface pool boiling condition was studied, which is included as Exp. no. 8 in the table. All conditions examined in this work are within the isolated bubble regime.

5.3.1 Upflow

Figure 5.1(a) depicts a photograph of a stream of vapor bubbles originating from a single nucleation site in upflow, for $G=258$ kg/m²-s, $\Delta T_{\text{sat}}=3.15^\circ\text{C}$ (Exp. no. 2b). Bubbles which emanate from the site are marked by the symbol "+" at the centroid. Shadows of the bubbles appear on the surface of the heater due to the lighting and camera angle. Arrows represent the direction in which the bubble is traveling. The behavior shown is typical of upflow nucleation at low heat flux. At the nucleation site, the bubbles experience a short

Table 5.1. Forced convective nucleation data.

Exp. no.	G ($\text{kg/m}^2\text{-s}$)	T_w ($^{\circ}\text{C}$)	ΔT_{sat} ($^{\circ}\text{C}$)	ΔT_{sub} ($^{\circ}\text{C}$)	J_a	q_w (kW/m^2)	h	$K \times 10^3$ (m/s^*)	n	d_b (mm)	d_L (mm)	τ_b (ms)	τ_L (N)
<i>upflow</i>													
1a	195	39.9	0.54	2.98	0.73	2.83	805	1.13	0.400	0.094	(98)	2.23	(92)
1b	192	39.8	2.39	2.83	3.23	4.80	920	2.46	0.463	0.165	(44)	22.2	(42)
1c	194	39.8	4.38	2.86	5.95	7.36	1017	4.07	0.500	0.207	(8)	4.67	(6)
2a	244	40.2	0.55	3.06	0.73	3.52	976	1.03	0.361	0.105	(110)	1.58	(106)
2b	258	40.2	3.15	3.27	4.28	6.92	1077	1.69	0.400	0.147	(83)	6.38	(79)
2c	255	40.0	6.34	2.78	8.56	10.9	1201	2.15	0.400	0.199	(53)	*	*
3a	315	40.3	1.32	2.52	1.74	3.63	945	1.26	0.419	0.112	(124)	2.23	(121)
3b	319	40.2	4.04	2.29	5.33	7.26	1147	1.58	0.408	0.160	(59)	1.31	(63)
3c	315	39.9	6.89	1.96	9.04	11.8	1338	2.05	0.404	0.204	(27)	*	*
<i>downflow†</i>													
4a	193	40.6	0.05	4.91	0.07	1.32	267	0.82	0.401	0.131 Δ	(4)	0.260	(4)
4b	192	40.6	4.60	4.28	6.43	4.78	538	1.23	0.374	0.157 Δ	(77)	0.324	(40)
4c	197	40.5	6.34	4.20	8.89	7.54	715	1.89	0.443	0.187 Δ	(31)	0.348	(20)
5a	246	40.8	2.95	4.78	4.15	3.24	418	1.29	0.413	0.134 Δ	(81)	0.254	(65)
5b	247	40.8	5.23	4.71	7.35	4.82	485	2.22	0.458	0.167 $\square$	(32)	0.286	(45)
5c	247	40.7	7.06	4.49	9.90	7.67	664	2.55	0.484	0.148 Δ	(59)	0.259	(56)
6a	315	41.0	3.18	5.00	4.45	3.93	469	1.00	0.384	0.150 $\square$	(20)	0.193	(34)
6b	313	41.1	5.08	4.84	7.09	5.04	495	2.05	0.495	0.112 $\square$	(73)	0.280	(32)
6c	311	41.1	6.30	7.79	8.79	6.66	583	2.59	0.500	0.132 $\square$	(50)	0.226	(37)
7	666	42.3	7.17	3.89	9.40	14.6	1319	2.91	0.488	0.123 ∇	(80)	0.186	(57)
<i>vertical pool boiling</i>													
8	0 (pool)	38.4	3.44	0.99	4.54	4.42	997	1.54	0.343	0.237	(34)	0.566	(34)
													9.36 (30)

* insufficient data due to "site coalescence"

† symbols represent bubble departure and sliding direction: Δ = upward; ∇ = downward; $\square$ = transitional, as described in Section 5.3.2

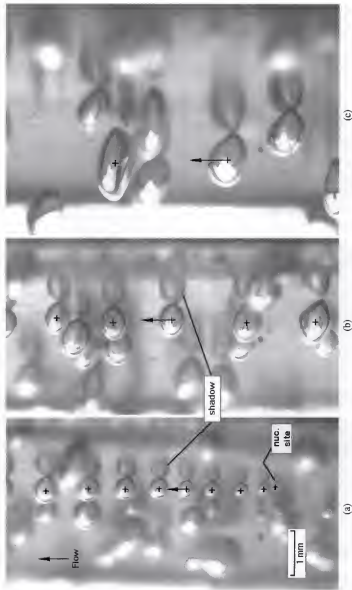


Figure 3. Photographs of bubbles originating from a single nucleation site in upflow, $G = 258 \text{ kg/m}^2\text{-s}$, $\Delta T_{\text{sat}} = 3.15 \text{ }^\circ\text{C}$ (Exp. no. 2b). (a) at the nucleation site. (b) approximately 10 mm downstream of site. (c) approximately 30 mm downstream of site.

period of stationary growth, and then depart from the site by sliding upward. Growth and departure is very regular, and the bubbles appear to be spherical in shape. The bubbles remain attached to the heating surface and continue to grow while sliding along the surface. Figures 5.1(b) and (c) depict two other streams of bubbles originating from the same nucleation site. The field of view is approximately 10 mm and 30 mm downstream of the site, respectively. As the bubbles become larger, they become slightly distorted, assuming a cap-like shape characteristic of bubbles rising faster than the surrounding flow. Bubbles are observed to oscillate laterally along the surface, most likely due to vortices shed by the bubbles, but generally remain attached to the heating surface. Bubbles that lift off the surface do so randomly, and tend to remain close to the heating surface. The fact that the bubbles do not typically lift off the surface is consistent with the supposition of Zeng et al. (1993) that in upflow boiling, the sliding bubble leads the surrounding flow, and the resulting shear lift force pushes the bubble against the wall, preventing lift-off.

Nucleation sites near the leading edge of the heater produced the majority of vapor bubbles, and very few sites were found downstream. Even at higher heat fluxes, the most active portion of the heater, in terms of the number of nucleation sites, was clearly the first 2 to 3 cm of the 30 cm long heater. The suppression of nucleation downstream is likely a result of several factors. First, the bubbles do not generally lift off of the surface, and the growth of the bubbles during sliding implies that energy is being depleted from the heating surface through latent heat transport. Also, the flow around the sliding bubbles may enhance bulk turbulent convection at the heater surface.

Collision and coalescence of vapor bubbles was observable at all but the lowest heat flux conditions. For example, at $G=244 \text{ kg/m}^2\text{-s}$ and $q''_w=3.52 \text{ kW/m}^2$ (Exp. no. 2a), nucleation sites were sparse and ebullition was well ordered. As the heat flux is increased

to $q''_w = 6.92 \text{ kW/m}^2$ (Exp. no. 2b), the nucleation site density increases, and bubbles from neighboring sites are occasionally observed to collide and coalesce. The waiting time also decreases. Upon further increasing the heat flux to $q''_w = 11.0 \text{ kW/m}^2$ (Exp. no. 2c), coalescence *at a nucleation site* becomes the dominant feature of the bubble dynamics. That is, the waiting time decreases to a point where a new bubble forms and grows at the nucleation site faster than the previous bubble can slide away, resulting in a collision and coalescence of the two bubbles. At even higher heat fluxes this waiting time becomes indistinguishably short, and a bubble may undergo repeated "site coalescence" before it is large enough to finally escape under the influence of buoyancy and/or drag.

At these higher heat fluxes, the flow near the surface becomes bubbly and chaotic, with neighboring bubbles colliding and numerous intermittent sites participating. Based on random fluctuations in the observed bubble trajectory, the flow near the wall appears to be highly turbulent. Under these conditions bubble lift-off was observed to occur on occasion. It appeared that a turbulent eddy or a passing bubble in the bulk liquid flow would randomly sweep the surface and lift off a bubble. Further downstream the entire flow field is bubbly and highly chaotic. Because lift-off occurred infrequently except during bubbly flow, and its occurrence appears to be random, lift-off diameters were not recorded for upflow.

5.3.2 Downflow

In downflow forced convection boiling, the buoyancy force experienced by a vapor bubble resists the quasi-steady drag exerted by the liquid flow. As a result, three bubble sliding conditions were observed, which depend largely on the local liquid velocity. These conditions are demonstrated in Figure 5.2, where photographs of bubble ebullition are shown at three different mass fluxes, $G = 246, 315, \text{ and } 666 \text{ kg/m}^2\text{-s}$ (Exp. nos. 5a, 6a, and 7). In Figure 5.2(a), the bubbles rise against the oncoming flow, behaving similar to that of upflow.

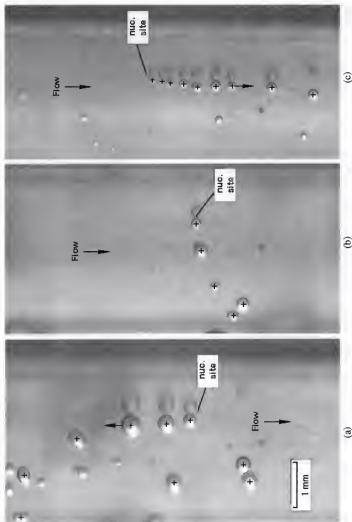


Figure 4. Photographs of bubbles originating from a nucleation site in downflow boiling. (a) $G=246 \text{ kg/m}^2\text{-s}$, $\Delta T_{\text{sat}} = 2.95^\circ\text{C}$ (Exp. no. 5a), (b) $G=315 \text{ kg/m}^2\text{-s}$, $\Delta T_{\text{sat}} = 3.18^\circ\text{C}$ (Exp. no. 6a), (c) $G=666 \text{ kg/m}^2\text{-s}$, $\Delta T_{\text{sat}} = 7.34^\circ\text{C}$ (Exp. no. 7).

The conditions under which this behavior is observed are marked with the symbol (Δ) in Table 5.1. However, bubbles do not slide far from the nucleation site before they are lifted off the surface into the bulk flow. Unlike in upflow, lift-off is not a result of random bulk or bubble-induced turbulence but is continuous and regular. This is consistent with the supposition of Zeng et al. (1993) that for the downflow configuration, the vapor bubble velocity will lag that of the surrounding fluid, and the resulting shear lift force should lift the bubble off the surface. As the liquid velocity is increased, the increased drag on a bubble overcomes the buoyancy force, and as shown in Figure 5.2(c), the bubble slides downward in the same direction as the bulk flow. This is marked in Table 1 by the symbol (∇). Lift-off occurs regularly, since the bubble still lags the surrounding flow. At elevated heat fluxes, bubble coalescence and eddies were observed in downflow as in upflow. But regardless of liquid velocity, the bubble sliding trajectory is much shorter in downflow compared with upflow due to lift-off.

As expected, there exists a transitional condition in downflow where bubble sliding was either upward or downward. This was observed in Exp. nos. 5b and 6a-c, where sliding took place with or against the flow with no preferred direction. These are marked in Table 1 with the symbol ($\square$). At times a bubble would simply remain attached to the site or remain within close proximity of the site. Figure 5.2(b) depicts a single photograph of a stream of vapor bubbles emanating from a nucleation site in which sliding is completely absent. The liquid velocity is such that sliding does not occur; it would appear that the buoyancy, drag, and growth forces balance. As a result, the bubble continues to grow until lifted off the surface, presumably by the shear lift force. Under these conditions smaller bubbles were often observed to grow in the wake of the larger coalesced bubble. It should be noted that,

while Figure 5.2(b) may represent the mean behavior under these conditions, the instantaneous behavior of the bubbles may resemble any of Figures 5.2(a)-(c).

In Section 5.3.1, it was suggested that bubble sliding is responsible to some extent for energy removal from the heating surface in vertical upflow. This point can be examined further by comparing the heat transfer rates for upflow and downflow, since in downflow boiling, bubble sliding is limited by the lift-off mechanism. Comparing the wall superheat in Table 5.1 for similar values of mass flux and heat flux, it appears that the wall superheat is generally lower for upflow than downflow boiling, indicative of a higher heat transfer rate. This point can be brought into clearer focus in Figure 5.3, where values of heat transfer coefficient, defined as $h = q_w / \Delta T_b$, are plotted for the upflow and downflow conditions of Table 5.1. From the figure it is clear that the heat transfer coefficient is greater for upflow than for downflow at otherwise similar flow and thermal conditions. It is reasonable to

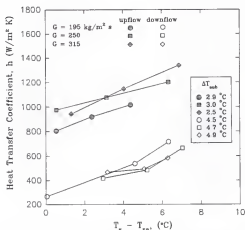


Figure 5.3. Plot of heat transfer coefficient, h , vs. wall superheat, $T_s - T_{sat}$, for the upflow and downflow conditions in Table 5.1.

conclude that bubble sliding is responsible for the higher heat transfer rates in upflow forced convection boiling. The importance and influence of bubble sliding on flow boiling heat transfer will be examined in detail in Chapter 7.

5.3.3 Vertical Pool Boiling

For the purpose of comparison, one boiling condition was examined under zero bulk flow conditions, which is listed as Exp. no. 20 in Table 1. A photograph of the ebullition process is presented in Figure 5.4. As in downflow, the bubbles regularly lift off of the surface. However, since there is no bulk fluid motion, the rising bubble experiences no significant velocity gradient from the surrounding fluid. As a result, there is no shear lift, and therefore another force must be responsible for lift-off. In Chapter 6, the hydrodynamic pressure force, previously examined by Klausner et al. (1993), is identified as the force responsible for vapor bubble lift-off in vertical pool boiling.

5.4 Bubble Dynamics Measurements

5.4.1 Growth Rate

Numerous investigators have attempted to model the growth of vapor bubbles in the simpler case of nucleate pool boiling. The most recent effort was made by Mei et al. (1995a&b), who performed a detailed numerical analysis to describe the growth of vapor bubbles in saturated pool boiling. They solved for the localized temperature variation within the heater surface as well as the liquid microlayer beneath the expanding bubble, and found that the growth rate is highly dependant on the rate of energy depletion from heater surface. By comparison, the modelling of flow boiling bubble growth is considerably more difficult for various reasons. In flow boiling, the bubble is subjected to a shear flow and is asymmetrically distorted. The thermal boundary layer surrounding the bubble is therefore

difficult to describe. Furthermore, when a bubble slides, the liquid microlayer is continually exposed to a different part of the heater surface, and the microlayer structure itself may change in response to the bubble's motion. Thus an accurate description of vapor bubble growth in flow boiling requires a detailed three-dimensional, three-phase heat and momentum transfer analysis. The purpose of this work is to collect experimental growth data, which is necessary to develop a better understanding of vapor bubble growth in order to provide a framework for future bubble growth models. Moreover, the experimental bubble growth data obtained in this investigation is necessary to estimate the forces on the growing bubble, which are used to develop the bubble dynamics model described in Chapter 6.



Figure 5.4. Photograph of bubbles originating from the same nucleation site in vertical pool boiling, $\Delta T_{\text{sat}} = 3.44^\circ\text{C}$ (Exp. no. 8).

A typical set of experimentally measured growth curves is depicted in Figure 5.5. A growth curve was obtained by measuring the diameter of an individual bubble from incipience through sliding. For saturated or nearly saturated boiling, the experimental growth rates may be curve-fit by a power law,

$$d(t) = K t^n \quad (5.1)$$

where $d(t)$ is the diameter of the bubble and t is the growth time. Because of the stochastic nature of the flow and thermal variations discussed in Section 5.2, a set of 5 growth curves were measured at various points in the film set, and the constants K and n were averaged to estimate the mean growth curve at this condition. Because in downflow the vapor bubbles resist the motion of the flow, the growth showed more variation, so a set of 10 growth curves were obtained at each condition. The averaged growth constants for upflow and downflow are included in Table 5.1. The Jacob number, Ja , is included in the tables for each condition. From the table it is observed that the growth data fit a power law ranging from about $t^{1/3}$ to $t^{1/2}$, which is expected for slightly subcooled boiling.

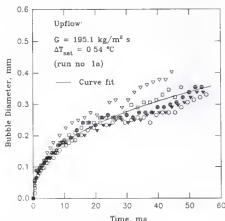


Figure 5.5. Typical bubble growth measurement.

In order to verify the statistical validity of the measured growth curves, additional growth data were obtained from the digital footage of ebullition for Exp. no. 1a. By increasing the total number of measured growth curves from 5 to 20, the resulting averaged constants K and n varied by less than 6 and 2 percent, respectively. Moreover, the scatter in the growth curves seen in Figure 5.5 was consistent, indicating that the variation in measured growth rate is primarily a result of statistical fluctuations in flow velocity and surface temperature, and not a result of a change in measurement accuracy. Therefore, a sample of 5 growth curves for upflow and 10 for downflow is judged acceptable to estimate the mean bubble growth curve.

A graphical comparison of the growth curves reveals useful information about the ebullition process. Figures 5.6(a) and (b) depict the mean growth curves for upflow and downflow, respectively. On both graphs, the number assigned to each curve represents the experiment number listed in Table 5.1. The solid triangles on both sets of curves represent the departure diameter at each condition. The upflow growth curves end approximately at the point where the vapor bubbles leave the field of view, except for curves 1c, 2b, 2c, 3b, and 3c, where bubble collisions and coalescence limited the available growth data. The downflow curves end when the bubbles lift off the heater surface, except for curve 4a, which extends beyond the limits of the plot.

Examining both graphs, several trends are clear. First, at a given mass flux, the bubble growth increases with increasing Ja (increasing ΔT_{sat}). Second, departure from the nucleation site occurs early in the growth curves, particularly in upflow, and therefore a large portion of bubble growth occurs during the sliding phase.

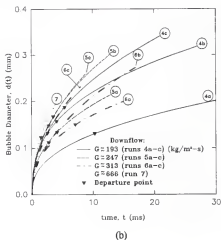
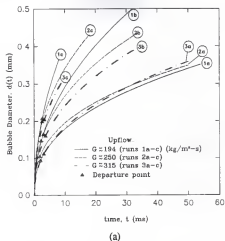


Figure 5.6. Comparison of mean growth curves; (a) upflow, (b) downflow. End of upflow curves represent limits of measuring diameter, due either to end of viewfield or coalescence. End of downflow curves represent mean lift-off point.

The effect of varying mass flux can be seen in both figures. In upflow it appears that, at low ΔT_{sat} (or Ja), the variation of mass flux has little impact on growth, as evidenced by curves 1a, 2a, and 3a in Figure 5.6(a), which represent conditions with roughly similar ΔT_{sat} (Ja). At higher wall superheat/Jacob number, the growth curves appear to separate, although this effect is less obvious since the wall superheat/Jacob number was not held constant. What is clear, however, is that the growth at these higher wall superheats/Jacob numbers *decreases* with *increasing* G. This is evident when comparing curves 1b, 2b, and 3b, and the curves 1c, 2c, and 3c, in which the growth decreases with increasing G despite a slight increase in ΔT_{sat} (Ja) in some cases. The same effect can be seen in Figure 5.6(b) for downflow, by examining the curves 4b, 5a, and 6a, and the curves 4c, 5b and 6b. These results suggest that the thermal boundary layer thickness is decreased with increasing G, which results in a greater amount of condensation from the dome of the bubble.

A final observation in the downflow growth curves deserves comment. At elevated ΔT_{sat} , as is the case for conditions 4c, 5c, and 7 in Figure 5.6(b), all with nearly the same value of wall superheat (approximately 7°C), the growth rates do not vary appreciably despite large changes in G (247 to $666 \text{ kg/m}^2\text{-s}$). Since n approaches 0.5, which is the maximum value for diffusion-controlled growth in saturated boiling, it can be concluded that microlayer evaporation is predominant and condensation is negligible.

5.4.2 Departure and Lift-off Diameter

The departure diameter, d_d , was measured from the film footage as the diameter of the vapor bubble in the frame immediately after the first sign of displacement from the nucleation site. Similarly, the lift-off diameter, d_L , is measured in the frame immediately after the bubble detaches from the heater surface. This method was also chosen by Klausner et al. (1993) in their measurement of departure and lift-off diameters. A source of

uncertainty in the measurement of bubble departure diameter arises because departure occurs early in the growth curve, when the bubbles are small and the diameter is increasing rapidly. Furthermore, bubble collisions, particularly at the nucleation site as described in Section 5.3.1, limited the available departure and lift-off diameter and waiting time data under some conditions. Mean values of d_d , d_L , and t_w at each experimental condition are included in Table 5.1. As discussed in Section 5.3.1, few bubbles were observed to lift off in upflow, and therefore no data for lift-off in upflow are presented.

Comparison of mean departure and lift-off diameter data. Figure 5.7 compares the mean values of departure diameter, d_d , at various values of mean liquid velocity as a function of wall superheat, ΔT_{sat} , for (a) upflow and (b) downflow. Also included in both plots is the mean departure diameter under vertical pool boiling, obtained in the same experimental facility under zero flow conditions. Each mean value of d_d and d_L results from a distribution of approximately 50 to 100 measurements; in the following section, the statistical relevance of the mean values will be examined. In upflow it is seen that the departure diameter clearly increases with increasing wall superheat; this is due to the increased bubble growth at higher wall superheat. The expanding vapor bubble generates a reaction force, referred to as the growth force, in the surrounding liquid which resists bubble motion away from the heater wall. While attached to the nucleation site, the vapor bubble is tilted due to the flowing liquid, and a component of the growth force parallel to the heater surface resists bubble departure. The larger the growth rate, the greater the resistance to departure, and therefore the larger the bubble will be before drag and/or buoyancy removes the bubble from the nucleation site. The departure diameter appears to decrease with increasing mean liquid velocity in upflow, although beyond $G = 250 \text{ kg/m}^2\text{-s}$ there is little change in d_d with G .

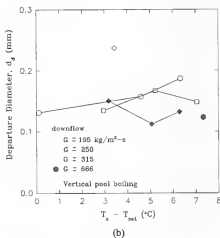
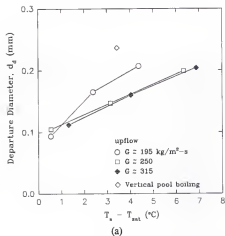


Figure 5.7. Comparison of measured mean departure diameters for (a) upflow and (b) downflow forced convection boiling. Diamond ($\diamond$) represents vertical pool boiling data point.

In downflow, interpretation of the departure diameter data is complicated by the fact that the bubbles depart either upward or downward. Under conditions where the bubbles slide upward against the flow, the departure diameter increases with increasing ΔT_{sat} , similar to upflow, although the values of d_d tend to be lower in downflow. On the other hand, under conditions which the vertical forces on the bubble were approximately in balance, and when the bubbles consistently departed downward with the flow, trends in the d_d data are less clear. Despite the differences in the data, it appears that, as in upflow, the departure diameter generally decreases with increasing mean liquid velocity.

Figure 5.8 depicts mean lift-off diameters obtained for downflow and vertical pool boiling at various mean liquid velocities. The lift-off diameter data follow similar behavior as departure diameter: in general d_L increases slightly with ΔT_{sat} and decreases with G . Both of these trends are expected. During the sliding phase, the bubble is upright, and the growth force acts entirely in a direction normal to the wall. An increase in ΔT_{sat} results in an increased growth rate, and therefore a larger growth force exists which resists lift-off. In addition the larger bubbles may contribute a larger surface tension force which helps maintain the bubble at the surface. Conversely, an increase in mass flux results in a higher shear lift force which promotes lift-off. Again, the variation of d_L with ΔT_{sat} is not discernable at $G = 250$ and $315 \text{ kg/m}^2\text{-s}$, due either to the transition from upward to downward sliding, or because lift-off typically occurs much later than departure, when the bubble growth rate and associated growth force is small. Therefore it is postulated that the growth force has a reduced influence on lift-off. It is also evident that the bubble lift-off diameter tends to decrease with increasing G . This is due largely to the increasing shear lift force.

Comparison of departure and lift-off diameter probability functions. To gain additional insight into the stochastic nature of the bubble removal process, probability

functions of departure and lift-off diameter were produced from selected experimental data. Strictly speaking, the experimental data samples are not large enough to create statistically valid probability functions, since such functions are subject to greater variation than the mean values. The functions are instead reported to demonstrate qualitative features of departure and lift-off behavior.

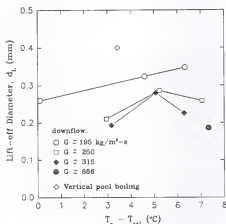


Figure 5.8. Comparison of measured mean lift-off diameters in downflow forced convection and vertical pool boiling.

Figures 5.9-5.11 depict probability functions of departure diameter obtained from the experimental data for 3 values of mean liquid velocity for upflow. Despite the small sample sizes, the distributions appear to be Gaussian although they are slightly skewed. From the figures it is evident that the distributions are distinct, and indicate that the mean values are statistically different. For upflow, the standard deviation of each distribution is consistently close to 1/10th the mean value, and the d_0 distribution broadens with increasing wall superheat. This result is similar to that reported by Klausner et al. (1993). The broadening of the d_0 distribution with increasing ΔT_{sat} is likely a result of increased wall temperature

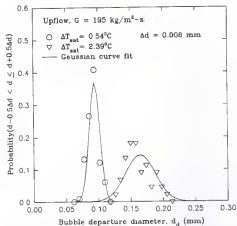


Figure 5.9. Departure diameter probability density functions at various wall superheats in upflow forced convection boiling; $G=195 \text{ kg/m}^2\text{-s}$.

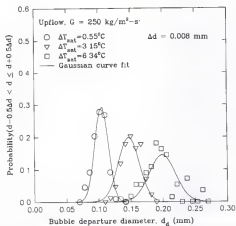


Figure 5.10. Departure diameter probability density functions at various wall superheats in upflow forced convection boiling; $G=250 \text{ kg/m}^2\text{-s}$.

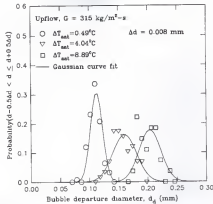


Figure 5.11. Departure diameter probability density functions at various wall superheats in upflow forced convection boiling: $G=315 \text{ kg/m}^2\text{-s}$.

variation at increased ΔT_{sat} . Using a liquid crystal thermography technique, Kenning (1991) measured statistical variations in wall superheat for saturated pool boiling of water. From his data it is evident that the wall temperature probability distribution broadens with increased ΔT_{sat} . Klausner et al. (1993) suggested that the origin of the stochastic behavior is the turbulent fluctuations of the liquid velocity, as well as spatial and temporal fluctuations of wall superheat which affect the bubble growth rate. Recently, Klausner et al. (1997) developed a model which predicts the stochastic variations of departure and lift-off diameter from estimates of the distribution of bulk liquid velocity and wall superheat. However, their model requires a method for predicting vapor bubble growth rate as a function of wall superheat, which at present is not available for vertical forced convection boiling with a slight degree of subcooling. Therefore no attempt has been made to apply their model to the present data.

Figures 5.12(a)-(b) depict probability functions of departure diameter for downflow. Again it appears that both the mean departure diameter and the standard deviation typically increase with increasing wall superheat. However, due to the varying direction of bubble departure the relative value of the standard deviation is less consistent than for upflow. Figures 5.13(a)-(b) depict probability functions of lift-off diameter for the same downflow conditions. For lift-off diameters, trends in the mean or standard deviation are less clear, although the standard deviation appears to decrease with increasing wall superheat.

5.4.3 Waiting Time

Waiting time, t_w , is defined as the time between bubble departure and the incipience of the next bubble at the same nucleation site. Mean waiting times are included in Table 5.1. Examining the tabular data, it appears that t_w tends to decrease with increasing heat flux. This is expected, since at higher heat flux the local temperature field beneath a nucleation site will recover more quickly than when exposed to a lower heat flux at otherwise identical conditions. At higher heat flux, the waiting times are so short that "site coalescence," described in Section 5.3.1, occurs readily at the nucleation site, and the waiting times under these conditions become too short to resolve within the temporal resolution of the film. Conditions under which waiting times could not be resolved are marked in Table 5.1 by asterisks (*).

From the film data it is possible to demonstrate a relationship between waiting time and departure diameter. Figure 5.14 illustrates a sequential plot of departure diameters and associated waiting times obtained from one vertical upflow boiling condition. From this plot it is evident that an increase in departure diameter is followed by a proportional increase in waiting time. This direct proportionality can be explained in terms of energy removal from the heater surface. A larger departure diameter implies that more energy has been consumed

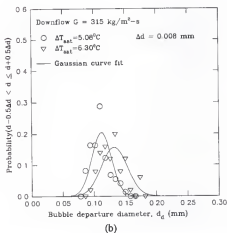
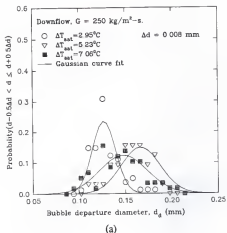
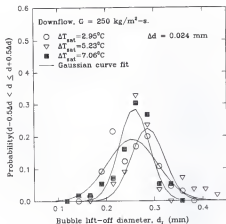
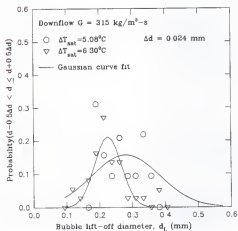


Figure 5.12. Departure diameter probability density functions for (a) $G=250 \text{ kg/m}^2\text{-s}$ and (b) $G=315 \text{ kg/m}^2\text{-s}$ at various wall superheats in downflow boiling.



(a)



(b)

Figure 5.13. Lift-off diameter probability density functions for (a) $G=250 \text{ kg/m}^2\text{-s}$ and (b) $G=315 \text{ kg/m}^2\text{-s}$ at various wall superheats in downflow.

from the nucleation site, and as a result the temperature field below the nucleation site takes longer to recover.

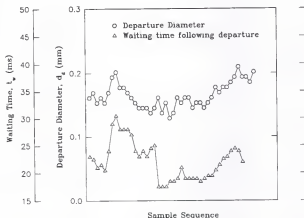


Figure 5.14. Sequence of measured departure diameters and associated waiting times for upflow boiling from the same nucleation site, $G=192 \text{ kg/m}^2\text{-s}$, $\Delta T_{\text{sat}}=2.39^\circ\text{C}$ (Exp. no. 1b). Sequence spans approximately 800 ms.

5.5 Discussion

Visual observations of ebullition in upflow and downflow forced convection boiling reveal several important features:

1. A relatively small portion of bubble growth occurs while the bubble is attached to the nucleation site; most growth occurs during the sliding process, particularly in upflow.
2. Vapor bubble lift-off is not generally observed in upflow, except occasionally due to random velocity fluctuations. In contrast, lift-off regularly occurs in downflow and thus vapor bubble sliding is not as dominant a heat transfer mechanism as in upflow.

3. Substantial evidence reveals that vapor bubble sliding enhances the energy transfer at the heating surface. First, the heat transfer coefficient is significantly higher for upflow than downflow at otherwise identical flow and thermal conditions. Second, in upflow boiling the most active nucleation sites were found close to the leading edge of the heater. Bubbles originating from those sites appeared to suppress nucleation downstream because sliding bubbles continue to remove energy from the surface through latent heat transport and enhanced bulk turbulent convection.

Growth rate, departure diameter, and lift-off diameter data have been obtained for upflow and downflow boiling in the isolated bubble regime for $G=192$ to $666 \text{ kg/m}^2\text{-s}$ and $q''_w=1.3$ to 14.6 kW/m^2 , at slightly subcooled conditions ($\Delta T_{\text{sub}}=1.0$ to 5.0°C). The data reveal a number of interesting features associated with the vapor bubble dynamics of vertical flow boiling:

1. Bubble growth increases with Jacob number (increasing ΔT_{sub}) under otherwise identical conditions in upflow or downflow. This is expected, since more energy is available at the nucleation site. On the other hand, bubble growth generally decreases with increasing mass flux. This is largely a result of operating at slightly subcooled conditions, and the increased mass flux provides increased condensation at the top of the bubble.
2. Bubble departure diameter increases with Jacob number (increasing ΔT_{sub}) in upflow and downflow, due to the increased growth force which retains the bubble at the nucleation site. The departure diameter generally decreases with increasing mass flux in upflow and downflow due to the increased quasi-steady drag force which pushes the bubble away from the site.

3. Lift-off diameters in downflow were observed to decrease with increasing mass flux, which is due to the greater shear lift imposed by the local liquid flow which removes the bubble from the surface. The lift-off diameter may increase with increasing Jacob number (increasing ΔT_{sat}), but this observation is not conclusive. This may be due to the reduced bubble growth force exerted on the bubble at lift-off, and therefore it is postulated that the growth force has a reduced effect on bubble lift-off.
4. Waiting time was observed to decrease with increasing heat flux, which is expected, since the energy at the heater surface near the nucleation site recovers more quickly after bubble departure at higher heat flux. In addition, waiting time and departure diameter are observed to be directly correlated, which is expected, since a larger bubble consumes more energy from the nucleation site, requiring more time for the site to recover.

CHAPTER 6 VAPOR BUBBLE DYNAMICS MODEL

6.1 Introduction and Literature Survey

Knowledge of the vapor bubble detachment process is essential to predicting the high heat transfer rates associated with boiling phenomena in the isolated bubble regime. For this reason many investigators have sought to understand the forces responsible for vapor bubble detachment. Significant progress has been recently made in understanding these forces in horizontal pool and flow boiling. However, despite the progress made over recent years, many issues remain unresolved.

In vertical flow boiling, the mechanisms of vapor bubble removal are uncertain, because in the absence of a gravitational force component normal to the heating surface, buoyancy no longer plays a major role in bubble lift-off. Zeng et al. (1993b) suggested several mechanisms which may influence the detachment of bubbles in vertical flow: (1) the shear lift force exerted on the bubble by the bulk flow influences the lift-off process by promoting lift-off in downflow and retarding lift-off in upflow; (2) during upflow, the sliding and continued growth of the bubble likely results in "a strong, unsteady, large scale, 3-dimensional two-phase flow," (Zeng et al., 1993b; p. 2279) and large scale transverse velocity fluctuations may be responsible for lift-off; (3) also during annular upflow, bubbles may simply merge into the vapor core.

The forces responsible for bubble departure and lift-off in flow boiling have been debated by a number of investigators. One of the first models of bubble departure diameter

in flow boiling was attempted by Chang (1963). His analysis assumed that departure occurs precisely at the point where the forces on the bubble balance. The forces considered were the buoyancy force, surface tension force, and dynamic forces normal and tangent to the heating surface. Sliding bubbles were not considered and the model was not verified. Levy (1967) developed a correlation for bubble departure in vertical, subcooled, upward flow boiling based on balancing the buoyancy, drag, and surface tension forces. He empirically deduced that the buoyancy force does not contribute significantly to bubble departure in upflow. Koumoutsos et al. (1968) measured bubble lift-off diameters from an artificial nucleation site in horizontal saturated flow boiling of water. Their model included three forces: drag, buoyancy, and surface tension. They experimentally observed sliding prior to lift-off at high liquid flow rates. Cooper et al. (1983) examined laminar upflow and downflow of n -hexane over a flat plate. The bubbles were observed to roll or slide away from the nucleation site, and they concluded that the departure point was not well defined. They also concluded that the simplified force balances used by Levy (1967) and Koumoutsos et al. (1968) were insufficient for describing their observed departure data.

Klausner et al. (1993) examined bubble departure and lift-off in horizontal flow boiling, and demonstrated that bubbles are inclined during growth due to the hydrodynamic drag force imposed by the flow. Their model incorporated the resulting asymmetric growth force, but required knowledge of the inclination angle as well as the advancing and receding contact angles. The contact area was also a required input. Zeng et al. (1993a&b) simplified Klausner's approach by assuming that the contact area of the bubble is so small at the point of departure that the resulting surface tension force is negligible. They developed a unified model for both pool and flow boiling, which simultaneously solves for the bubble diameter and inclination angle at the point of departure. The model compares very well to all known

experimental data reported in the literature, and to date is the most accurate model available for predicting horizontal pool and flow boiling departure diameters for the isolated bubble regime.

Other investigators have since examined the departure process as well. van Helden (1994) and van Helden et al. (1994) examined forces on growing bubbles in vertical upflow, and concluded that buoyancy, lift, drag and surface tension are all important. To examine the relative magnitude of the surface tension force, they assumed the bubble shape to be purely hemispherical, with the base attached directly to the wall. This ignores the presence of the microlayer, and thus their analysis overestimates the size of the contact area and the influence of the surface tension force. They did not propose a bubble detachment model. Kandlikar and Stumm (1995) obtained departure data in horizontal forced convection boiling of water, and concluded that Zeng's (1993b) model for departure diameter was effective only for diameters greater than 500 μm , since in their judgement surface tension forces cannot be neglected for smaller bubbles. They proposed a control-volume approach to examine separately the forces on the upstream and downstream portions of the bubble, and neglected inertial forces resulting from the growth of the bubble. Their control volume approach fails to conserve momentum, and no prediction for departure diameter was proposed. Finally, Klausner et al. (1997) considered the stochastic features of vapor bubble departure and lift-off diameters, and developed a mechanistic model to predict the probability density functions (pdf's) of departure and lift-off diameters in flow boiling. They demonstrated that the stochastic behavior of departure and lift-off diameters is controlled by stochastic variations in wall superheat and mean liquid velocity.

In this investigation, a predictive model for vapor bubble sliding trajectories and departure and lift-off diameters in vertical upflow and downflow forced convection boiling

is presented. The primary motivation for this work is to gain insight into the bubble dynamic process; specifically, what forces govern the detachment and motion of vapor bubbles in flow boiling. An analysis of the forces on a growing and sliding bubble is based on the work by Klausner et al. (1993) and Zeng et al. (1993b) for horizontal flow boiling, and is expanded to account for the acceleration of the sliding bubble. The model is compared with experimental bubble sliding trajectory data, then with experimental departure and lift-off diameter data obtained in Chapter 5 for single-phase inlet flow boiling. The model is also compared with the departure and lift-off data for horizontal stratified flow reported by Klausner et al. (1993) and Zeng et al. (1993b). An analysis of the relative influence of the governing forces is presented, and the results of the comparison are discussed.

6.2 Development of Model

The development of the present model follows the framework set forth by Zeng et al. (1993b) for horizontal flow boiling bubble departure and lift-off. However the observations reported in Chapter 5 for the case of vertical flow boiling requires a reexamination of many of the governing assumptions in their model. This work deals with a wide range of operating conditions, including upflow, downflow, and vertical pool boiling. The degree of influence that a given force has on the vapor bubble dynamics can change with the operating condition. Furthermore, the model will be extended to predict vapor bubble sliding trajectories. The governing assumptions and improvements to the model are discussed in the development which follows.

Figure 6.1 illustrates a vapor bubble growing while (a) attached to its nucleation site, and (b) sliding along the heater surface in the presence of vertical upflow. The x- and y-momentum equations which describe the departure, sliding, and lift-off processes may be expressed as:

$$\sum F_x = F_b + F_{qt} + F_{du,x} + F_{s,x} + F_{fs,x} + F_{am,x} = \rho_v V_b \frac{du_{bc,x}}{dt} \quad (6.1a)$$

$$\sum F_y = F_{du,y} + F_{sl} + F_{s,y} + F_h + F_{cp} + F_{fs,y} + F_{am,y} = \rho_v V_b \frac{du_{bc,y}}{dt} \quad (6.1b)$$

where F_b is the buoyancy force, F_{qt} is the quasi-steady drag in the flow direction, F_{du} is the growth force, F_s is the surface tension force, F_{fs} is the free stream acceleration force, F_{am} is the added mass force, F_{sl} is the shear lift force, F_h is the hydrodynamic lift force, and F_{cp} is the contact pressure force. On the right hand side of (6.1a) and (6.1b), u_{bc} is the velocity of the bubble at its center of mass, V_b is the bubble volume, and ρ_v is the vapor density. A complete description of each term in Equations (6.1a) and (6.1b) is presented below.

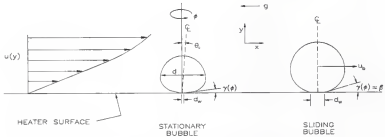


Figure 6.1. Illustration of vapor bubble stationary and sliding growth.

6.2.1 Buoyancy Force

Following Zeng et al. (1993b), the buoyancy force is estimated by

$$F_b = \frac{4}{3} \pi a^3 (\rho_\ell - \rho_v) g \quad (6.2)$$

where a is the bubble radius and g is the gravitational acceleration. For the system depicted in Figure 6.1, F_b acts in the positive x -direction, through the centroid of the bubble.

In Equation (6.2) and through much of the remaining theoretical development, the shape of the vapor bubble is approximated to be a perfect sphere. The validity of this approximation is examined as follows. First, examining the images obtained from the experimental work, examples of which are presented in Chapter 5, the vapor bubbles indeed appear to maintain an approximately spherical shape throughout the measured stationary and sliding periods. Furthermore, the bubble Reynolds number, $Re_b = 2\Delta Ua/\nu_\ell$, ranged from 30 to 300; the bubble Weber number, $We_b = \rho_\ell \Delta U^2 2a/\sigma_\ell$, ranged from 10^{-3} to 10^{-2} . For such values of Weber number, it is expected that little distortion of the vapor bubble takes place during the measured growth period, and the bubble remains spherical. It should be noted, however, that in vertical upflow the bubbles remain attached to the heater surface, and hydrodynamic distortion of the vapor bubbles is observed further downstream of the nucleation site, typically 3 to 4 hydraulic diameters. It is expected that when the Weber number exceeds approximately unity, distortion of the bubble is likely to take place. Beyond that point the effectiveness of the present model is uncertain, although the qualitative results of the present work remain intact.

6.2.2 Quasi-Steady Drag and Shear Lift Forces

To obtain an estimate for the quasi-steady drag force, Mei and Klausner (1992) performed a numerical analysis of a spherical bubble moving in an unbounded liquid flow. Based on their results, Mei et al. (1994) developed the following expression to estimate the quasi-steady drag, a similar version of that used by Zeng et al. (1993 a and b):

$$F_{gr} = 6(C_w)\pi\rho_\ell\nu_\ell\Delta Ua\left\{\frac{2}{3} + \left[\left(\frac{12}{Re_b}\right) + 0.75\left(1 + \frac{3.315}{Re_b^{1/2}}\right)^{-1}\right]\right\}, \quad (6.3)$$

where $\Delta U = u_{bc} - u_\ell$ is the relative velocity between the bubble and the liquid at the bubble center of mass, $Re_b = 2\Delta Ua/\nu_\ell$ is the bubble Reynolds number, ν_ℓ is the liquid kinematic

viscosity, and C_w is a wall correction coefficient adopted in this study. The direction in which F_w acts depends on the relative velocity, but is assumed to act entirely in the x -direction, through the bubble centroid. Previous analyses leading to the above expression, with $C_w=1$, neglect the presence of the wall, which may result in error. To account for such an effect, the empirical constant C_w is included, and was determined by comparing the experimental bubble sliding trajectories obtained from upflow, downflow, and vertical pool boiling with that predicted by the model. A value of $C_w = 2.0$ was found to correlate the experimental bubble trajectory well over the entire range of flow and thermal conditions which were measured in this work.

To estimate the shear lift force, Zeng et al. (1993b) used the result of Mei and Klausner (1994) for a bubble exposed to an unbounded shear flow. Recently, however, Legendre and Magnaudet (1997) revealed an error in their result at low Reynolds number, and reported a corrected version of the shear lift force for creeping flow. An approximation for the shear lift force, presented in the Appendix B, is obtained by interpolation using the result of Legendre and Magnaudet for small Reynolds number and the result of Auton (1987) for large Reynolds number:

$$F_{SL} = -\frac{1}{2} \rho_f |\Delta U| \Delta U \pi a^2 \alpha^{1/2} \left\{ \left[\frac{1.146 J(\sqrt{2\alpha/Re_b})}{Re_b^{1/2}} \right]^2 + \left(\frac{3}{4} \alpha^{1/2} \right)^2 \right\}^{1/2} \quad (6.4)$$

where

$$\alpha = \left| \frac{dU}{dy} \frac{a}{\Delta U} \right|$$

$$J(\epsilon) = 0.6765 \{1 + \tanh [2.5(\log_{10} \epsilon + 0.191)]\} \{0.667 + \tanh [6(\epsilon - 0.32)]\}$$

F_{SL} acts on the bubble centroid in either the positive or negative y -direction, depending on the relative velocity, $\Delta U = u_{bc,x} - u_t$. If $\Delta U > 0$, the shear lift force acts in the negative y -

direction. Conversely, if $\Delta U < 0$, a positive shear lift results. As an approximation, the force is assumed to act entirely in the y-direction, even during stationary growth when the bubble is tilted.

In Equations (6.3) and (6.4), the liquid velocity profile near the wall is estimated using Reichardt's single-phase turbulent flow expression

$$\frac{U(y)}{u^*} = \frac{1}{\kappa} \ln \left(1 + \kappa \frac{yu^*}{v_\ell} \right) + c \left[1 - \exp \left(-\frac{yu^*/v_\ell}{\chi} \right) - \frac{yu^*/v_\ell}{\chi} \exp \left(-0.33 \frac{yu^*}{v_\ell} \right) \right] \quad (6.5)$$

where $\kappa = 0.4$, $\chi = 11$, and $c = 7.4$. Using the definition of friction factor, the friction velocity, u^* , can be expressed as

$$u^* = \sqrt{\frac{\tau_w}{\rho_\ell}} = u_\ell \sqrt{C_f/2} \quad (6.6)$$

where u_ℓ is the mean liquid velocity. For single phase turbulent flow which is encountered in this work, Petukhov's formula is used to estimate the friction factor, C_f :

$$C_f/2 = (2.236 \ln Re_\ell - 4.639)^{-2} \quad (6.7)$$

where Re_ℓ is the bulk liquid Reynolds number, $Re_\ell = GD_b/\mu_\ell$, and D_b is the hydraulic diameter of the duct.

6.2.3 Growth Force

Zeng et al. (1993b) used a modified form of an expression developed by Klausner et al. (1993) to estimate the growth force based on a hemispherical bubble expanding in an inviscid liquid,

$$F_{dx} = -\rho_\ell \pi a^2 \left(\frac{3}{2} C_s \dot{a}^2 + a \ddot{a} \right) \quad (6.8)$$

where (*) indicates differentiation with respect to time. C_s is an empirical constant; a value of C_s equal to 20/3 was found to correlate nearly all departure data in pool boiling, and was

used successfully for horizontal flow boiling. The same value is used in the present analysis. Values of a , $\bar{a}$, and $\bar{a}$ are taken from mean growth curves which were experimentally obtained in Chapter 5, and are given in terms of bubble diameter in power-law form,

$$d(t) = Kt^n \quad (6.9)$$

where K and n are measured constants. During the stationary growth phase (when the bubble is attached to the nucleation site), Klausner et al. (1993) demonstrated that the bubble grows asymmetrically, which they modelled as a bubble growing at an inclined angle θ_i to the normal direction. Thus the growth force acts along the centerline of the bubble depicted in Figure 6.1, with respective x- and y-direction components given by

$$F_{dx} = F_{du} \sin \theta_i \quad \text{and} \quad F_{dy} = F_{du} \cos \theta_i \quad (6.10)$$

6.2.4 Surface Tension Force

The surface tension forces in the x- and y-directions are given by

$$F_{s,x} = - \int_0^{2\pi} (d_w / 2) \sigma \cos \gamma \cos \phi \, d\phi \quad (6.11)$$

$$F_{s,y} = - \int_0^{2\pi} (d_w / 2) \sigma \sin \gamma \, d\phi \quad (6.12)$$

where d_w is the contact diameter of the bubble with the wall, σ is the surface tension, ϕ is the angle about the line normal to the contact surface as shown in Figure 6.1, and $\gamma(\phi)$ is the general contact angle. At a given angle ϕ , the local surface tension force acts on the point of contact between the bubble surface and the solid surface, along the contact angle $\gamma(\phi)$. As implied by Equation (6.11), when the microlayer wedge angle is uniform, the x-direction component of this force is symmetric about the centerline, and the resulting net surface

tension force in the x-direction is zero. Zeng et al. (1993b) argued that both the x- and y-direction components are negligible because the presence of the microlayer beneath a growing bubble results in a small value of d_w . Indeed, a computational study of pool boiling by Lee and Nydahl (1989) indicates that the surface tension force is an order of magnitude less than the buoyancy and growth forces near the departure point. In vertical flow the component of the gravitational force normal to the surface is zero, but the surface tension force is still much less than the growth force. For this reason the surface tension force is neglected in determining departure diameter. However, during the sliding phase the growth and subsequent growth force is reduced, and the bubble becomes less distorted and more spherical as shown in Figure 6.1. The exact shape of the microlayer could not be measured, but it is likely that the bubble microlayer wedge angle increases as the bubble takes on a more spherical shape, resulting in an increased contact diameter, d_w . The larger wedge angle and contact diameter imply that the surface tension force gains importance in the lift-off process. The surface tension force is therefore estimated in the following manner. The bubble is assumed to slide in an upright position, thus the x-direction of the surface tension force is zero by (6.12), and so the force acts entirely in the y-direction. Because the bubble is assumed to grow symmetrically, the microlayer wedge angle γ is assumed to be uniform and is approximated as the receding liquid/solid contact angle, β . The receding liquid/solid contact angle for FC-87 and nichrome was measured in the laboratory and found to be approximately 6.5° . Modeling the sliding bubble as a truncated sphere, and using a straightforward geometric analysis, the contact diameter d_w is approximated as $d_w = 2a \sin \beta$. Equation (6.12) then reduces to $F_{s,y} = -d_w \sigma \pi \sin \beta$.

6.2.5 Contact Pressure and Hydrodynamic Lift Forces

The contact pressure force results from the pressure difference between the inside and outside of the bubble over the contact area of the bubble. The hydrodynamic force is estimated by considering inviscid flow over a truncated bubble, which is truncated due to contact with the heater surface; due to the symmetry of the flow over the majority of the bubble surface, the contribution to F_x is from the pressure on top of the bubble over the bubble contact area.

Zeng et al. (1993a) showed that the contact pressure force, F_{cp} , may be neglected relative to the surface tension force, and the hydrodynamic lift force, F_h , may in turn be neglected relative to the shear lift force. This was also found to be true in this work, with one exception. For the case of vertical pool boiling, the absence of bulk flow implies that no shear lift exists. Under these conditions the hydrodynamic lift force, which occurs during sliding, may play an important role. Therefore the hydrodynamic lift is included in modelling lift-off, which was estimated by Klausner et al. (1993) as

$$F_h = \frac{9}{8} \rho_t \Delta U^2 \frac{\pi d_w^2}{4} \quad , \quad (6.13)$$

where ΔU is evaluated at the center of mass of the bubble.

6.2.6 Free-Stream Acceleration and Added Mass Forces

The free-stream acceleration and added mass forces arise due to unsteady motion of the bubble and surrounding liquid. The free-stream acceleration force, F_a (Mei, 1996), is

$$F_a = \frac{4\pi}{3} \rho_f a^3 \frac{D\vec{U}}{Dt} \quad ,$$

where $\vec{U}$ is the fluid velocity vector. The second force is the added-mass force, estimated by Mei (1996) on a spherical bubble as

$$F_{am} = \frac{2\pi}{3} \rho_f a^3 \left(\frac{D\mathcal{U}}{Dt} - \frac{d\mathcal{V}}{dt} \right)$$

where $\mathcal{V}$ is the bubble velocity vector. In the above, $\frac{D}{Dt} = \frac{\partial}{\partial t} + u_\ell \cdot \nabla$ is the fluid material derivative. Because this work is concerned with predicting mean bubble behavior, the stochastic features of the liquid velocity are ignored in the present model. For this reason F_b is neglected, and assuming the bubble's motion to be entirely in the x-direction, F_{am} reduces to

$$F_{am} = -\frac{2}{3}\pi a^3 \rho_\ell \frac{du_{bc,x}}{dt} \quad (6.14)$$

where $u_{bc,x}$ is the velocity of the bubble's center of mass. The motion of the bubble center of mass in the y-direction (due solely to growth) is neglected in calculating the added-mass force, for two reasons. First, the motion of the bubble center of mass away from the wall is small, and the y-component of the growth force is dominant. Second, the departure and lift-off model of Zeng et al. (1993 a and b) neglects the added-mass force, so the added-mass effect, though small, is accounted for in the empirical constant in Equation (6.8).

It should be noted that Zeng et al. (1993b) also neglected the added-mass force in predicting lift-off diameters in horizontal flow boiling. This is because they assumed that the dominant forces controlling lift-off in horizontal flow boiling are the buoyancy and growth forces, which do not require knowledge of the bubble velocity. Therefore no attempt was made to predict bubble sliding trajectory. The validity of this assumption will be examined in Section 4.3. In vertical flow boiling, however, the shear lift force plays a significant role in lift-off, which requires a reliable estimate of the bubble velocity.

6.2.7 Departure Model

Applying the above approximations and assumptions, the following equations are used to estimate the mean bubble diameter and inclination angle at the point of departure in vertical forced convection and pool boiling:

$$\sum F_x = F_b + F_{gs} + F_{du}\sin\theta_i = 0 \quad (6.15a)$$

$$\sum F_y = F_{du}\cos\theta_i + F_{SL} = 0 \quad (6.15b)$$

The acceleration of the bubble center of mass while the bubble is attached to the nucleation site is neglected in these relations. Equations (6.15a) and (6.15b) are identical to the model of Zeng et al. (1993b) except for the inclusion of the empirical constant C_w in Equation (6.3) and the fact that the buoyancy force acts in the x-direction. The simultaneous solution of Equations (6.15a) and (6.15b) yields predictions of departure diameter, d_d , and inclination angle, θ_i .

6.2.8 Sliding Trajectory and Lift-off Model

Using the above approximations for the sliding phase of bubble growth, the following expressions are solved to estimate the bubble trajectory and lift-off diameter:

$$\sum F_x = F_b + F_{gs} = \left(\rho_v V_b + \frac{1}{2} \rho_l V_b \right) \frac{du_{bc,x}}{dt} \quad (6.16a)$$

$$\sum F_y = F_{du} + F_{SL} + F_{s,y} + F_h = 0 \quad (6.16b)$$

where the added mass force has been included on the right side of (6.16a), and the acceleration of the bubble center of mass in the y-direction is neglected. The above equations are solved using a time-marching scheme beginning at the point of departure. The acceleration term, $du_{bc,x}/dt$, is approximated by a second-order finite difference. Using the

experimental growth rate curves and the force relations, the displacement x from the nucleation site is calculated at each time step using Equation (6.16a). The forces in the y -direction are also calculated, and lift-off is taken to be the point at which they balance, i.e. when $\Sigma F_y = 0$.

6.3 Results

6.3.1 Comparison of Model with Experimental Data

Figures 6.2-6.6 compare typical experimental bubble sliding trajectories with the results of the present model for vertical pool boiling, as well as for upflow and downflow forced convection boiling. The empirical constant C_w , used in estimating the quasi-steady drag force, was set to a value of 2.0 in order to obtain a best fit to all experimental bubble trajectory data. It is seen that, despite the empiricism of a single value of C_w , the model compares well to experimental bubble trajectories in both vertical pool and flow boiling. Moreover, the model accurately captures the change in bubble sliding direction which occurs in downflow under increasing flow velocity. The model thereby suggests that the dominant forces controlling bubble motion along the wall in vertical flow and pool boiling are the buoyancy and the quasi-steady drag.

The predicted departure and lift-off diameters are included in Table 6.1 for comparison with experimental data. Figure 6.7(a) compares the measured and predicted departure diameters for the upflow, downflow, and vertical pool boiling conditions in this work, along with departure radii reported by van Helden (1994) for vertical upflow and pool boiling of water at an artificial cavity (diameter 0.119 mm) on a platinum heating surface. The comparison is very good over the range of flow conditions studied in this work;

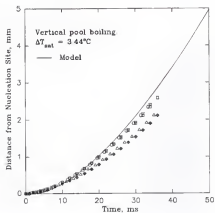


Figure 6.2. Comparison of measured and predicted bubble sliding trajectories: vertical pool boiling.

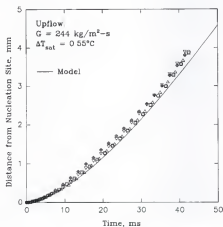


Figure 6.3. Comparison of measured and predicted bubble sliding trajectories: upflow.

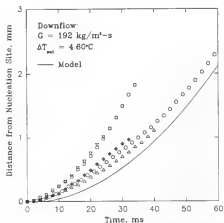


Figure 6.4. Comparison of measured and predicted bubble sliding trajectories: downflow, $u_f = 0.112 \text{ m/s}$.

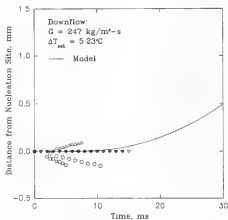


Figure 6.5. Comparison of measured and predicted bubble sliding trajectories: downflow, $u_f = 0.144 \text{ m/s}$.

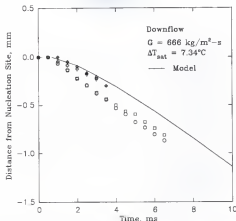


Figure 6.6. Comparison of measured and predicted bubble sliding trajectories: downflow, $u_t = 0.390$ m/s.

however, the model significantly under-predicts the departure data of van Helden. The reason for this is that the size of the artificial cavity is not negligible compared with the measured bubble departure diameters. Since the contact diameter, d_w , is at least equal to the cavity diameter, the assumption that the surface tension force is negligible at departure is not valid. This is significant because it demonstrates that the cavity size can have a large effect on the departure diameter and therefore on the energy removal from the nucleation site.

To characterize the performance of the model with the present data, the relative deviation is used, which is defined as

$$r.d. = \frac{\sum_{k=1}^N \frac{|d_{meas,k} - d_{pred,k}|}{d_{meas,k}}}{N} \times 100$$

where N is the number of data points, and the subscripts 'meas' and 'pred' refer to the respective measured and predicted diameters. Neglecting the measurements of van Helden

(1994), for the data shown in Figure 6.7(a), $r.d.=20.5\%$, which is acceptable considering the range of flow and thermal conditions examined.

Table 6.1. Forced convective nucleation data.

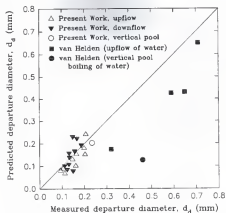
Exp No.	G (kg/m ² -s)	u_t (m/s)	ΔT_{sat} (°C)	q''_w (kW/m ²)	$d_{L,meas}$ (mm)	$d_{L,pred}$ (mm)	$d_{L,meas}$ (mm)	$d_{L,pred}$ (mm)
<i>upflow</i>								
1a	195	0.114	0.54	2.83	0.094 Δ	0.082		
1b	192	0.112	2.39	4.80	0.165 Δ	0.154		
1c	194	0.113	4.38	7.36	0.207 $\Delta^{\dagger}$	0.243		
2a	244	0.143	0.55	3.52	0.105 Δ	0.090		
2b	258	0.151	3.15	6.92	0.147 Δ	0.131		
2c	255	0.149	6.34	10.9	0.199 Δ	0.181		
3a	315	0.184	1.32	3.63	0.112 Δ	0.069		
3b	319	0.187	4.04	7.26	0.160 Δ	0.103		
3c	315	0.185	6.89	11.8	0.204 Δ	0.154		
<i>downflow</i>								
4a	193	0.112	0.05	1.32	0.131 $\Delta^{\dagger}$	0.109	0.260 †	0.299
4b	192	0.112	4.60	4.78	0.157 Δ	0.167	0.324	0.307
4c	197	0.115	6.34	7.54	0.187 $\Delta^{\dagger}$	0.194	0.348 †	0.317
5a	246	0.143	2.95	3.24	0.134 Δ	0.140	0.254	0.279
5b	247	0.144	5.23	4.82	0.167 $\square$	0.223	0.286	0.303
5c	247	0.144	7.06	7.67	0.148 Δ	0.231	0.259	0.308
6a	315	0.183	3.18	3.93	0.150 $\square^{\dagger}$	0.079	0.193 †	0.250
6b	313	0.182	5.08	5.04	0.112 $\square$	0.102	0.280	0.262
6c	311	0.181	6.30	6.66	0.132 $\square$	0.158	0.226	0.283
7	666	0.390	7.17	14.6	0.123 ∇	0.086	0.186	0.209
<i>vertical pool boiling</i>								
8	0	0	3.44	4.42	0.237 Δ	0.204	0.566	0.524

† insufficient data to construct pdf's

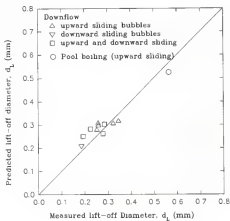
Δ bubbles typically depart and slide upward

∇ bubbles typically depart and slide downward

$\square$ bubbles depart and slide upward or downward from the nucleation site, and at times do not slide at all



(a)



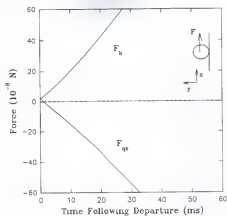
(b)

Figure 6.7. Comparison of predicted and measured (a) departure and (b) lift-off diameters for vertical pool and flow boiling.

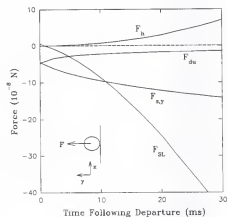
Figure 6.7(b) compares the measured and predicted values of lift-off diameter for downflow and pool boiling. The model predicts that lift-off will not occur for upflow, which is consistent with observations. The predicted lift-off diameters compare well with the measured values over a range of bubble dynamic behavior. This is further elucidated by separating the data into three regimes: (1) upward bubble departure and sliding, as is the case for low u_f , (2) downward bubble departure and sliding, which occurs at high u_f , and (3) bubble departure and sliding upward or downward, with no preferred direction on average. Despite such complex conditions the model compares well with the experimental data. The relative deviation for the data shown in Figure 6.7(b) is 13.2%. The fact that the relative deviation for lift-off diameter is smaller than that of departure diameter may be because the growth force has a reduced influence on lift-off. Since the growth force is controlled by the wall superheat, the reduced growth force implies that stochastic variations in wall superheat have a reduced effect on the lift-off behavior.

6.3.2 Examination of Contributing Forces in Sliding and Lift-off Model

It is instructive to examine the relative contribution of the forces which contribute to the sliding and lift-off processes at the varying flow conditions examined in this work. Figures 6.8(a) and (b) compare the respective x- and y-direction forces predicted by the model for a sliding vapor bubble in vertical upflow, $u_f=0.151$ m/s, $\Delta T_{\text{sat}}=3.15^\circ\text{C}$ (Experiment no. 2b in Table 6.1). In the x-direction it is seen that the quasi-steady drag force is initially positive, indicating that the upward flow velocity is leading the bubble velocity. But the drag force quickly becomes negative as the bubble overtakes the liquid due to buoyancy and leads the bulk flow. In the y-direction, the dominant force is the negative shear lift force, which behaves in a similar manner to the quasi-steady drag force. The shear lift force, and to a lesser extent the surface tension force, are clearly the dominating forces



(a) x-direction



(b) y-direction

Figure 6.8. Comparison of forces predicted by model on sliding vapor bubble: *upflow*, $u_f = 0.151$ m/s, $\Delta T_{sat} = 4.30^\circ\text{C}$.

which retain the bubble at the heater surface in upflow. The hydrodynamic lift force is of minimal importance, and it is also seen that the growth force becomes vanishingly small during sliding. This lends validity to the approximation that the microlayer is less pronounced, which was assumed in estimating the surface tension force.

Figure 6.9(a) and (b) depict x- and y-direction forces for vertical downflow, $u_f = 0.143$ m/s, $\Delta T_{\text{sat}} = 2.95^\circ\text{C}$ (Experiment no. 5a in Table 6.1). In this case the quasi-steady drag always acts in the negative x-direction since the bubble always lags the downward flow velocity. In the y-direction it is clear that the dominant forces are the shear lift and surface tension forces, and it is the shear lift overtaking the surface tension that produces lift-off. As in upflow, both the growth and hydrodynamic lift forces are small.

Figures 6.10(a) and (b) compare the x-and y-direction forces predicted by the model for vertical pool boiling, $\Delta T_{\text{sat}} = 3.44^\circ\text{C}$ (Experiment no. 8 in Table 6.1). In this case the quasi-steady drag is initially zero, and increases in the negative x-direction as the bubble accelerates upward in the quiescent liquid. No shear lift is present, and the growth force again diminishes over time. But surprisingly, the hydrodynamic lift force is considerably larger, since the relative velocity between the bubble and the surrounding liquid is much greater. Eventually it is the hydrodynamic lift which overcomes the surface tension to produce lift-off.

6.3.3 Application of Model to Horizontal Flow Boiling

In order to gain further insight into the vapor bubble dynamics in horizontal flow boiling, the present model is applied to the departure and lift-off data of Klausner et al. (1993) and Zeng et al. (1993b), obtained from horizontal stratified flow boiling of R-113. Neither bubble growth rates nor sliding trajectories were measured. Since no model for flow boiling growth is available, the growth rates are approximated using Zuber's (1961)

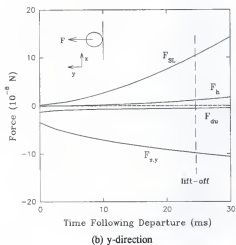
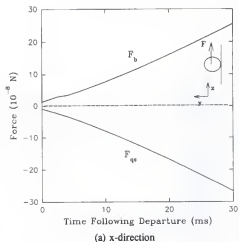


Figure 6.9. Comparison of forces predicted by model on sliding vapor bubble: *downflow*, $u_f = 0.143$ m/s, $\Delta T_{sat} \approx 5.01^\circ\text{C}$.

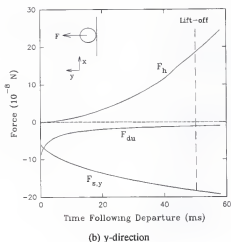
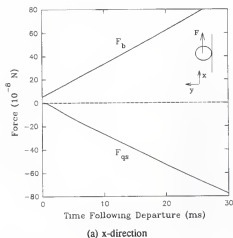


Figure 6.10. Comparison of forces predicted by model on sliding vapor bubble: *vertical pool boiling*, $\Delta T_{\text{sat}} = 4.53^\circ\text{C}$.

diffusion-controlled bubble growth solution for saturated pool boiling under one-g sub-atmospheric and atmospheric conditions,

$$a(t) = \frac{2b}{\sqrt{\pi}} Ja \sqrt{(\eta t)} \quad (6.17)$$

where $Ja = \rho_l C_{p,l} \Delta T_{sat} / \rho_v h_{fg}$ is the Jacob number, η is the liquid thermal diffusivity, $C_{p,l}$ is the liquid specific heat, h_{fg} is the latent heat of vaporization, and b is an empirical constant which is supposed to account for asphericity of the bubble. It is seen that Zuber's correlation predicts the bubble growth to be proportional to $t^{1/2}$; in fact, the actual time dependance may be lower, say, $t^{1/4} \leq t \leq t^{1/2}$, due to depletion of energy at the nucleation site. Regardless, Zeng et al. (1993b) found the Zuber correlation to give satisfactory results by adjusting the constant b to provide a best fit of the departure and lift-off model to their experimental data. Using the model of Zeng et al., incorporating the corrected shear lift force estimate given in Equation (6.4), a value of $b = 0.9$ is found to correlate the departure and lift-off diameters. This value is adopted for the present analysis as well. To estimate the quasi-steady drag, a value of $C_w = 2.0$ in Equation (6.3) was assumed. This value is consistent with the model applied to vertical flow investigated previously. Finally, the receding liquid-solid contact angle for R-113 and nichrome was measured in the laboratory to be approximately 8.4° . This value was used to calculate the surface tension force during sliding.

Figures 6.11(a) and (b) compare the respective departure and lift-off diameters for horizontal flow boiling with those predicted by the present model. The comparison is good, with relative deviations of 22.3 and 13.8 percent for the departure and lift-off diameters, respectively. This is quite acceptable, considering the use of the Zuber relation in approximating the bubble growth rates. Furthermore, though not depicted, the model of

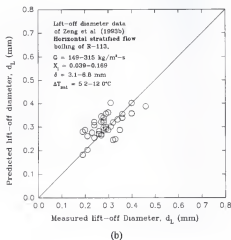
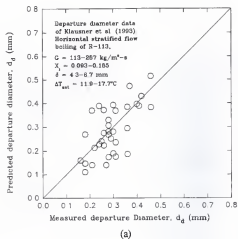


Figure 6.11. Comparison of measured and predicted departure and lift-off diameters for horizontal stratified flow boiling of R113.

Zeng et al. (1993b) results in relative deviations of 20.0 and 18.6 percent for the respective departure and lift-off diameters. It should be noted that the calculation of departure diameter is relatively insensitive to the value of C_w in the expression for the quasi-steady drag force. Since this is the only major difference between the present departure model and that of Zeng et al. (1993b), the accuracy of both models are similar.

On the other hand, the present model improves the prediction of lift-off diameters by 26 percent. The reason for this improvement can be seen by comparing the two models as follows. Figures 6.12(a) and (b) depict the respective x- and y-direction forces participating in sliding and lift-off in horizontal flow boiling predicted by the present model for one set of data, $u_t = 0.82$ m/s, $\delta = 4.1$ mm, and $\Delta T_{\text{sat}} = 5.2^\circ\text{C}$. It is seen that the buoyancy and growth forces contribute to some extent in the lift-off process. However, the model also predicts that shear lift and surface tension are important and may at times dominate the lift-off process. This is in contrast to the model of Zeng et al. (1993b), which assumes the shear lift and surface tension forces to be negligible.

Further evidence of the importance of the shear lift force is revealed by reexamining the experimental lift-off data. Figure 6.13 depicts three sets of lift-off diameter data reported by Zeng et al. (1993b) at approximately the same wall superheat and varying values of liquid phase velocity. The model of Zeng et al. (1993b) predicts that only the buoyancy and growth forces affect lift-off, which are not dependent on bulk liquid velocity. However, the plot clearly shows that the lift-off diameter tends to decrease with increasing u_t , suggesting the increasing influence of a positive shear lift. The trend is indeed captured by the present model, as evidenced by the predictive curves included in Figure 6.13.

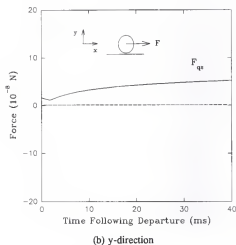
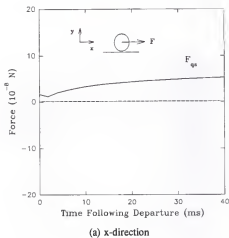


Figure 6.12. Comparison of forces predicted by model on sliding vapor bubble, *horizontal stratified flow boiling*, $u_t = 0.82$ m/s, $\delta = 4.1$ mm, $\Delta T_{sat} = 5.2^\circ\text{C}$.

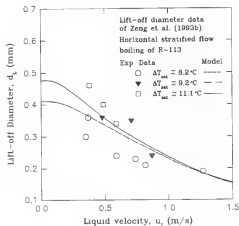


Figure 6.13. Variation of lift-off diameter in horizontal stratified flow with liquid velocity; comparison with results of present model.

6.4 Discussion

A predictive model for bubble sliding trajectories and departure and lift-off diameters has been developed for single-phase inlet vertical upflow and downflow forced convection and vertical pool boiling. The model accurately captures the sliding trajectories and dynamic behavior of vapor bubbles over a wide range of vertical flow and pool boiling conditions, and compares well to experimental departure and lift-off diameter data, as well as that reported in the literature for horizontal flow boiling.

The analysis presented in this chapter is restricted to the isolated bubble boiling regime, with the governing flow conditions being single-phase liquid flow. A number of additional assumptions and approximations are made as necessary to estimate the forces on the vapor bubble. The purpose of this development is not to develop a universal model of bubble detachment, but rather to gain insight into the governing mechanisms which influence

vapor bubble detachment and motion in flow boiling. Despite the approximations, the model is demonstrated to compare well with a range of flow boiling data, and supports many features of flow boiling ebullition which have been experimentally observed.

An analysis of the model reveals several important features of the bubble dynamics of vertical and horizontal flows, and identifies several key forces which play a role in bubble sliding and lift-off:

1. The sliding velocity of the bubbles in upflow, downflow, and vertical pool boiling is dominated by buoyancy and quasi-steady drag forces.
2. The dominant forces controlling lift-off in vertical flow boiling are the shear lift and surface tension forces. A strong negative shear lift force acts with surface tension to prevent lift-off in upflow, while in downflow a positive shear lift force overcomes the surface tension to promote lift-off.
3. In vertical pool boiling a large hydrodynamic lift force is responsible for overtaking the surface tension force and causing lift-off.
4. In horizontal flow boiling, sliding is controlled by the quasi-steady drag. Lift-off is controlled to a lesser extent by the buoyancy and bubble growth forces, and to a greater extent by shear lift and surface tension. This in contrast to the model of Zeng et al. (1993b), which assumed shear lift and surface tension to be negligible.

CHAPTER 7 THE VAPOR BUBBLE SLIDING MECHANISM

7.1 Introduction and Literature Survey

The development of reliable mechanistic boiling heat transfer models requires an understanding of the complex heat transport mechanisms and their mutual interaction. In forced convection boiling, this understanding is further complicated by the fact that vapor bubbles not only remove energy from the heating surface while attached to nucleation sites, but can also potentially remove energy following departure.

Mesler (1988) suggested that the high heat fluxes associated with annular flow boiling are due to secondary nucleation. According to Mesler, secondary nucleation is the process whereby an expanding vapor bubble ruptures a thin liquid film, thus causing the film to entrain vapor bubbles as it recedes. The supposed entrained vapor bubbles serve as nucleation centers from which bubble growth proceeds within the superheated liquid film. Mesler's hypothesis was not confirmed by Zeng and Klausner (1993), who conducted a photographic investigation of horizontal flow boiling in thin films. Thorncroft and Klausner (1997) studied high-speed video images of vertical upflow and downflow boiling in the annular flow regime and did not observe secondary nucleation as described by Mesler.

In Chapter 5, a detailed investigation is reported of the bubble dynamics in slightly subcooled vertical upflow and downflow forced convection boiling of FC-87. The experiments, which were primarily performed in the isolated bubble regime, reveal a marked difference in the behavior of the bubbles in upflow and downflow. In upflow, vapor bubbles

are observed to slide away from the nucleation site in the direction of the flow. The bubbles typically do not lift off the heating surface, except due to occasional local turbulent fluctuations. During sliding, the bubbles continue to grow and remove energy from the heating surface. At low superheat ($\Delta T_{sat} < 4.0^\circ\text{C}$), the majority of vapor bubbles emanate from a few nucleation sites located near the leading edge of the heater. It would appear that the sliding bubbles remove sufficient energy from the heater surface to suppress nucleation downstream. In contrast, during downflow the vapor bubbles regularly lift off the heating surface within close proximity of the nucleation site. Nucleation sites were scattered over the entire surface, and the bubbles would slide only a short time before being lifted off. The difference in the bubble lift-off behavior is attributed to a shear lift force exerted by the liquid near the wall. In upflow, a sliding bubble leads the local flow, resulting in a shear lift force which pushes the bubble toward the surface. In downflow, a bubble will lag the local flow, which results in a shear lift force which expels the bubble from the surface.

A comparison of the heat transfer coefficients for slightly-subcooled ($3\text{--}5^\circ\text{C}$) upflow and downflow boiling with single-phase inlet conditions are also reported in Chapter 5, in Figure 5.3. It is clear that, for the flow and thermal conditions encompassing the isolated bubble regime, the heat transfer coefficient is significantly greater in upflow than in downflow under otherwise identical conditions. It was postulated that the difference in vapor bubble dynamics was responsible for this heat transfer enhancement; specifically, sliding bubble heat transport is considerably more prevalent in upflow since bubbles do not detach from the heating surface.

The focus of this investigation is to provide experimental evidence to support the hypothesis that sliding bubble heat transport can be significant in flow boiling. The consequences of such a finding are of fundamental importance. It suggests that there does

exist a secondary heat transfer mechanism in flow boiling, but is due to sliding bubbles and not secondary nucleation. Furthermore, it suggests that the conventional practice of using pool boiling correlations for modeling flow boiling heat transfer is not fundamentally sound. Moreover, in addition to quantifying the nucleation site density, waiting time, vapor bubble growth rate, and departure diameter specifically for flow boiling, an additional parameter, the lift-off diameter, must be quantified in order to mechanistically model flow boiling heat transfer.

The bubble sliding process was observed as early as 5 decades ago by Gunther (1951), but only recently has it received much critical attention. Cornwell (1990) measured the heat flux from a tube in a bundle of tubes exposed to bubbly flow of saturated water. He observed that, in the presence of bubbly flow, the heat flux was more than double that of single-phase turbulent convection. He proposed that the increase was due to the sliding of vapor bubbles along the surface of the tube. Moreover, he noted that the upstream tubes were entirely responsible for nucleation, and that "liquid convection and the influence of the sliding bubbles account for the total heat transfer" in the downstream tubes. Recently, Houston and Cornwell (1996) performed similar tests using air bubbles injected into the tube bundle to demonstrate that turbulence enhancement caused by the sliding bubbles is as important as microlayer evaporation to the energy removal from the tubes.

Tsung-Chang and Bankoff (1990) presented a theoretical study which demonstrates the importance of microlayer evaporation in highly subcooled nucleate flow boiling heat transfer. They proposed that the sliding of vapor bubbles would increase the microlayer evaporation by a factor of two or more over a stationary bubble.

Yan and Kenning (1996) used liquid crystal thermography to examine the temperature distribution on an inclined plate in a pool of water subjected to a sliding vapor bubble. They

reported that the local heat transfer coefficient in the wake of a sliding vapor bubble was approximately three times that for single-phase natural convection ahead of the bubble. Using air bubbles injected into the flow, they demonstrated that the liquid flow around the bubble creates a localized region of greatly enhanced heat transfer in the wake behind the bubble. Yan et al. (1996) have shown that for small bubbles the majority of heat transfer takes place in the bubble's wake, and the heat transfer contribution due to latent heat transport is significant only for large bubbles.

All of the aforementioned studies have acknowledged the importance of microlayer evaporation energy transport beneath the bubble, but only a few recent studies have sought to resolve the heat transfer contribution due to enhanced bulk turbulence resulting from the sliding bubbles. The purpose of this investigation is therefore twofold: (a) to demonstrate the relevance of the sliding bubble energy transport mechanism to the total heat transfer in forced convection boiling, and (b) to resolve whether enhanced latent heat transport or bulk turbulence is primarily responsible for the heat transfer due to vapor bubble sliding.

7.2 Results

7.2.1 Examination of Macroscale Heat Transfer

Isolated bubble boiling regime. Regarding the heat transfer measurements depicted in Figure 5.3, an experiment was performed to further highlight the difference in bubble sliding as it contributes to the macroscale heat transfer in upflow and downflow boiling. The experimental facility was operated to attain single-phase, slightly subcooled (approximately 4°C) flow at the entrance of the test section. Power was applied to the test section to achieve vigorous boiling at the heater surface, then reduced until only isolated sites were active. This operating procedure eliminates the effect of boiling hysteresis. Measurements of thermal

and hydrodynamic properties of the flow were recorded; simultaneously, the number of active nucleation sites was estimated by visual count and recorded for each value of heat flux. For reference, measurements of single-phase (non-boiling) heat transfer were made at each condition. The measured single-phase heat transfer coefficients are in good agreement with Petukhov's turbulent heat transfer correlation.

Figure 7.1 compares the ratio of the measured two-phase heat transfer coefficient, h , to the single-phase (non-boiling) heat transfer coefficient, $h_{i,s}$, for upflow and downflow as a function of the measured nucleation site density. Several trends are apparent: First, $h/h_{i,s}$ increases as the number of active nucleation sites increases for both upflow and downflow, as expected. However, the heat transfer coefficient increases *at a much greater rate* in upflow than downflow. This implies that a significantly smaller number of active sites are required to achieve the same heat transfer in upflow as would be achieved in downflow under the same hydrodynamic conditions. In other words, an active nucleation site in upflow results in greater heat transfer from the surface than in downflow. It is noted that both $h/h_{i,s}$ curves appear to converge as $n/A \rightarrow 0$, which is expected, since single-phase flow heat transfer is not dependant upon flow orientation. This result provides strong evidence that there exists a significant heat transfer mechanism related to the sliding of vapor bubbles.

Extension to other flow boiling regimes. The difference in macroscale heat transfer between upflow and downflow forced convection boiling is not limited to the isolated bubble regime. Figure 7.2 compares the heat transfer coefficients for upflow and downflow measured for one set of single-phase flow inlet conditions. The heat transfer coefficient is consistently higher for upflow than downflow over a range of heat flux which extends from the isolated bubble regime to vigorous boiling.

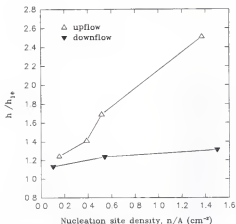


Figure 7.1. Heat transfer coefficient ratio, $h/h_{1\phi}$, vs. nucleation site density for upflow and downflow boiling in the isolated bubble regime; $G=255 \text{ kg/m}^2\text{s}$, $\Delta T_{\text{sat}} \approx 5^\circ\text{C}$.

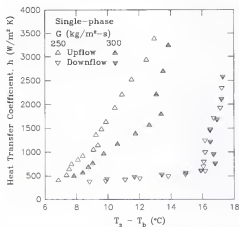


Figure 7.2. Comparison of boiling curves for single-phase inlet upflow and downflow.

It is noted that a distinct change in slope is apparent in the h vs. ΔT_b curve for downflow at $\Delta T_b \approx 17^\circ\text{C}$; the increase in slope was observed to correspond with the formation of large (approximately 3-4 mm) bullet-shaped bubbles on the heater surface for large values of ΔT_b . These large bubbles were seen to move about the surface, coalescing with smaller bubbles and either remaining somewhat static, or sliding upward against the flow. This behavior is consistent with the hypothesis of Wadekar and Kenning (1994) that such static bubbles enhance heat transport by acting as heat pipes, providing a conduit for latent heat transport from the wall directly to the subcooled bulk liquid. However, this behavior was observed only in downflow boiling with single-phase inlet conditions; it was not observed in upflow, since the combined action of drag and buoyancy sweeps the bubbles along the heater surface.

Figure 7.3 displays the heat transfer coefficients for annular upflow and downflow boiling under otherwise similar conditions. Static bubbles seen in downflow boiling were not observed in either annular upflow or downflow boiling. The higher liquid velocity for annular flow sweeps the bubbles downstream in both upflow and downflow and prevents the formation of larger bubbles. Moreover, the increased liquid velocity and turbulence causes bubbles to slide and lift off similarly in both annular upflow and downflow, and the differences in vapor bubble dynamics between the two conditions were not easily distinguishable. As a result, the h vs. ΔT_b curves for annular upflow and downflow are similar, as demonstrated. This does not imply that the vapor bubble sliding mechanism is not important in annular flow boiling; it merely implies that the bubble sliding heat transfer mechanisms are similar for upflow and downflow under these specific conditions.

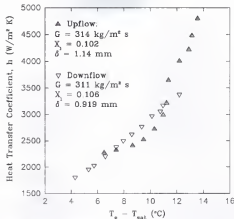


Figure 7.3. Comparison of heat transfer coefficient vs. wall superheat for annular upflow and downflow boiling.

7.2.2 Estimation of Sliding Heat Transfer

In order to obtain additional evidence and understanding of sliding bubble heat transfer, the following experiment was conducted. Under vertical upflow conditions, vigorous boiling was established on the heating surface of the visual test section; then the heat flux was reduced until only approximately 10 nucleation sites were active. Attempts were made to activate only one nucleation site, but this could not be accomplished. Using the high-speed motion video camera, the growth and initial sliding phase of the bubbles from one site was photographed; from this video the growth, departure diameter, and streamwise sliding trajectory were measured. Meanwhile the nucleation site density, as well as measurements of flow rate, surface temperature, and heat flux were recorded.

The bubble sliding contribution to the total heat transfer was estimated as follows. Following Cornwell (1990), the total heat transfer rate per unit area of heater is approximated as

$$q''_{\text{tot}} = q''_{1\phi} + q''_I + q''_{II} \quad , \quad (7.1)$$

where $q''_{1\phi}$ is the heat flux due to single-phase convection, q''_I is the heat flux due to bubble growth at the nucleation site (i.e. excluding sliding), and q''_{II} is the heat flux due to bubbles growing and sliding while attached to the heater surface. Experiments show that the rate of heat transfer for single-phase turbulent flow is described well by Petukhov's correlation, which is used in the above analysis. It is noted that q''_{II} represents the combined effect of latent heat transport and enhanced turbulent convection due to bubble agitation of the thermal boundary layer.

The average latent heat transfer rate from a single nucleation site (not including the sliding contribution) can be described as

$$\overline{q''_{n, \text{site}}} = \frac{\rho_v h_{fg} (4/3) \pi (d_d/2)^3}{t_d + t_w} \quad , \quad (7.2)$$

where ρ_v is the vapor density, h_{fg} is the latent heat of vaporization, d_d is the diameter of the vapor bubble at departure, t_d is the time of departure, and t_w is the waiting time, defined as the time between the departure of the bubble and the incipience of the next bubble at the same site. The total latent heat transfer rate due to stationary bubble growth (i.e., not including sliding) is then

$$q''_I = \left(\frac{n}{A} \right) \overline{q''_{n, \text{site}}} \quad , \quad (7.3)$$

where (n/A) is the nucleation site density.

For the following experimental conditions, the measured data from the flow boiling facility are as follows:

Experimental Conditions:

- $G = 242 \text{ kg/m}^2\text{-s}$
- $\Delta T_{\text{sub}} = 1.2^\circ\text{C}$
- $\Delta T_b = 2.2^\circ\text{C}$
- $q_w'' = 1.41 \text{ kW/m}^2$
- $A_s = 3.9 \times 10^{-3} \text{ m}^2$

Measurements:

- $d(t) = (2.87 \times 10^{-3})t^{0.5} \text{ m}$
- $d_b = 0.124 \text{ mm}$
- $t_d = 3.9 \text{ ms}$, $t_w = 17.0 \text{ ms}$
- Approx. 10 active sites

Analysis of these data shows that the sliding bubble contribution to the total convective heat transfer is approximately:

$$\begin{aligned} q_{sl}'' &= q_{\text{tot}}'' - q_{nb}'' - q_f'' \\ &= 1.41 - 0.68 - 0.14 \text{ (kW/m}^2\text{)} \\ &= 0.59 \text{ kW/m}^2 \end{aligned}$$

Thus the heat transfer contribution due to sliding bubbles is approximately 42% of the total heat transfer under these conditions.

7.2.3 Sliding Air Bubble Heat Transfer

The heat transfer associated with vapor bubble sliding contains contributions from both enhanced turbulent convection and latent heat transport, which could not be distinguished in the preceding experiments. In an attempt to distinguish the enhanced turbulent convection contribution from that due to latent heat transport, air bubbles were injected along the heating surface as follows. Using the air bubble injection system described in Section 2.3, single-phase upflow was established in the test section, and air was injected at the surface of the heater to allow the air bubbles to attach to the heater surface. Single-phase upflow was chosen as the operating flow condition in order to promote bubble sliding along the heater

surface. Air flow rates were chosen such that, at the minimum air injection rate, the conditions resembled nucleation from a single isolated nucleation site, and at the maximum air injection rate, the flow appeared bubbly. At a given air injection rate, power to the test section heater was increased, in increments beginning at zero, and measurements of heat transfer coefficient and flow conditions were recorded. The heater power was increased in small increments to delay the onset of heterogeneous boiling.

Figure 7.4 compares the total heat transfer coefficient, h , as a function of bulk temperature difference, $\Delta T_b = T_s - T_b$, for single-phase flow and air bubble injection rates: 7.5 ml/min, 17.0 ml/min, and 27.0 ml/min. As expected, the heat transfer coefficient during single-phase flow without air injection is essentially constant over the range of bulk temperature difference until nucleate boiling is observed. With air bubble injection, h also remains essentially constant with ΔT_b , although the magnitude of h is much larger than that of single-phase flow, and increases as the air flow rate increases. The fact that the heat transfer coefficient does not change with ΔT_b for a given air injection rate suggests that the bulk turbulent transport is enhanced by the motion of the air bubbles, resulting in greater heat transport at the wall.

Further evidence of this enhanced bulk turbulent transport can be seen in Figure 7.5, which compares heat transfer coefficient profiles for the same air flow rate at different values of liquid subcooling, $\Delta T_{sub} = T_{sat} - T_b$. It is seen that, over a range of liquid subcooling (3.5° to 10.1°C), the heat transfer coefficient profiles remain approximately constant as ΔT_b is increased, until nucleate boiling is observed. Beyond this point the curves diverge, which is expected, since the onset of heterogeneous boiling occurs at increasingly higher ΔT_b as the liquid subcooling is increased.

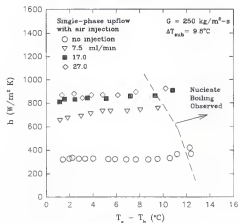


Figure 7.4. Measured heat transfer coefficient vs bulk temperature difference for single-phase inlet upflow under various air bubble injection rates.

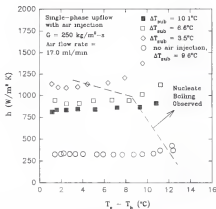


Figure 7.5. Measured heat transfer coefficient vs. bulk temperature difference for single-phase inlet upflow with air bubble injection under varied liquid subcooling.

A trend observed in Figure 7.5 which is difficult to explain is that the heat transfer rates for $\Delta T_{\text{sub}}=3.5^{\circ}\text{C}$ are slightly greater than those for $\Delta T_{\text{sub}}=10.1$ and 6.6°C . In the absence of evaporation, it is expected that the degree of subcooling should not influence the bulk turbulent heat transport associated with sliding vapor bubbles. Thus the possibility of evaporation at the liquid/air interface of the bubbles cannot be discounted for the $\Delta T_{\text{sub}}=3.5^{\circ}\text{C}$ case. Further investigations are required to conclusively resolve the understanding of this trend.

7.3 Discussion

The experiments presented herein provide conclusive evidence demonstrating the enhancement of heat transfer due to sliding bubbles in forced convection boiling. The distinct differences noted in upflow and downflow boiling heat transfer are directly attributable to the different bubble sliding trajectories for the different flow configurations. In upflow boiling with single-phase flow inlet conditions, it was found that sliding bubble heat transfer enhancement can account for as much as 40 percent of the total energy transfer.

The air bubble injection experiments reported in this work demonstrate that bulk turbulence enhancement is a significant mechanism in the heat transfer in forced convection boiling. This result is in agreement with the findings of Yan et al. (1996) and Houston and Cornwell (1996). In fact, for the experimental conditions considered in this work, it appears that bulk turbulent enhancement accounts for the majority of heat transfer enhancement due to sliding bubbles.

It has been well known that both bulk turbulent convection and nucleate boiling contribute significantly to forced convection boiling heat transfer. The typical modeling practice has been to use a method of superposition and simply add the bulk turbulent heat

transfer contribution to that of nucleate boiling. However, the validity of this approach must be reexamined, since it assumes independence of the bulk turbulent and nucleate boiling heat transfer mechanisms. Considering the bubble sliding heat transfer enhancement data presented, it may be concluded that the bulk turbulent and nucleate boiling mechanisms are in fact interdependent.

CHAPTER 8

CONCLUSIONS AND RECOMMENDATIONS

In this work, a series of studies were undertaken to examine the fundamental, small-scale behavior of forced convection boiling and its effect on the macroscale heat transfer. Coincident with the experimental and analytical study, the soundness of the Rohsenow-Chen superposition method of predicting flow boiling heat transfer rates has been considered. This chapter lists the important accomplishments and findings of this research, and concludes with recommendations for future research.

8.1 Accomplishments and Findings

The research presented in this document has increased the fundamental knowledge and understanding of forced convection boiling heat transfer as follows.

1. A fundamental analysis of the parameters which govern the active nucleation site density has resulted in the introduction of a new dimensionless parameter, the cavity size ratio, which has been shown to correlate nucleation site density. The cavity size ratio has also been shown to be a leading order dimensionless parameter on which the suppression of nucleation sites depends. This work has culminated in the introduction of a flow boiling regime map, which discriminates the purely convective regime from the nucleate boiling regime. The significance of this work is that, by separating flow boiling heat transfer data into two characteristic regimes, future models of heat transfer can be developed which predict each regime separately, resulting in improved reliability. Fu and Klausner (1997) used this regime map in the development of a separated flow heat transfer model for the purely

convective regime. Using the flow boiling regime map, they were able to isolate a wide range of convective regime data from the flow boiling data reported in the literature. Prior to the present work, there was insufficient data available to judge the reliability of separated flow heat transfer modeling.

2. An experimental and photographic study of upflow and downflow boiling has yielded data and knowledge on vapor bubble growth rate, departure diameter, sliding trajectory, lift-off diameter, and waiting time in the isolated bubble regime. The vapor bubble growth rate, departure and lift-off diameters, and the sliding trajectory have been observed to be dependant upon the bulk liquid flow rate, indicating that the ebullition process and bulk turbulence are interdependent. The waiting time and departure diameter are also observed to be co-dependent, demonstrating that each are related to the energy depletion from the nucleation site. In addition, the bubble dynamics, particularly the lift-off behavior, in upflow and downflow were observed to be fundamentally different. In upflow, vapor bubbles typically remain attached to the heater surface due to a negative shear lift force; in downflow, a positive shear lift force is responsible for lifting bubbles off the heating surface in a regular manner. As a result, the sliding bubbles in downflow participate to a much lesser degree in the enhanced heat transfer associated with bubble sliding. Indeed, measurements of heat transfer rates in upflow and downflow reveal significantly higher heat transfer rates in upflow than in downflow under the same wall superheat and flow conditions.

3. Using the improved knowledge of the ebullition process obtained from experiments, a model of vapor bubble detachment and motion has been developed for vertical upflow and downflow boiling. The model compares well to departure diameter, sliding trajectory, and lift-off diameter data obtained from the experimental work presented herein as well as in the literature. The model also performs well when applied to the horizontal

flow boiling data of Klausner et al. (1993) and Zeng et al. (1993b). The present model in fact is more accurate in predicting lift-off diameter than the original model proposed by Zeng et al. (1993b). Moreover, the present model predicts that the lift-off diameter is velocity-dependent, a fact that is evident in the vapor bubble data of Zeng et al. (1993b) but not captured by their model.

4. An experimental study of vapor bubble sliding has demonstrated that the sliding mechanism contributes significantly to the macroscale heat transfer. Experiments incorporating air bubble injection have shown that enhanced turbulent convection is at least as important as latent heat transport to the vapor bubble sliding mechanism. Both of these results are very important because they demonstrate that the heat transfer mechanisms in forced convection boiling, namely bulk turbulent convection and ebullition, are not simply additive effects, but are in fact interdependent. This raises concern as to the validity of the Rohsenow-Chen superposition technique of modeling flow boiling heat transfer.

8.2 Recommendations for Future Research

The present work has been undertaken with the goal of understanding the mechanisms governing flow boiling heat transfer so that a fundamental, mechanistic heat transfer model may be formulated. Several key issues remain unresolved, and are recommended for future research:

1. The development of a reliable vapor bubble growth model remains to be developed for flow boiling. This would likely begin with the growth model developed by Mei et al. (1995a and b) for pool boiling, and incorporate the vapor bubble dynamic model developed in this work. The model must necessarily capture the effect of the local turbulent convection on bubble growth, which at present is not well understood.

2. Expansion of the knowledge and prediction of active nucleation site density is required for flow boiling. This requires further scrutiny of the effect of bulk turbulence, as well as a better understanding of the role of surface characteristics of the heater. One suggested study involves micro-machining an array of artificial cavities on the heating surface. The experimental observation of the nucleation at these sites would lead to a greater understanding of which site geometries and sizes act as nucleation sites, and under what flow and thermal conditions these sites become active.

3. The development of a microgravity bubble dynamic model is important to the space industry. In space-based applications of heat transfer, forced convection boiling is not utilized, because the absence of a gravitational field adds uncertainty to the vapor removal process, and whether forced convection boiling can be sustained in microgravity remains unresolved. For this reason, only single-phase closed-loop heat exchangers are utilized in space-based convective heat transfer systems. On the other hand, two-phase heat exchangers would be smaller, lighter, and more efficient. Therefore taking advantage of nucleate flow boiling heat transfer could free up costly space and reduce weight which could be used for additional payloads. Application of the present vapor bubble dynamic model to microgravity conditions will demonstrate whether or not there exists a vapor removal mechanism, and therefore suggest whether microgravity flow boiling can be sustained. Measurements of vapor bubble detachment diameters in microgravity would be useful to evaluate the model.

4. Measurement of the characteristics of a turbulent flow boundary layer in the presence of sliding bubbles is needed in order to further understand the bulk turbulence/ebullition interaction. One suggested experiment involves measurement of velocity components of flow over solid spheres which are attached to a wall. Further investigations might seek to measure velocity fluctuations of adiabatic, single-phase flow with air bubble

injection at the wall. Both of these studies may lead to improved prediction of the turbulent thermal and hydrodynamic boundary layer profiles near the heating surface.

APPENDIX A PROPERTIES OF FC-87

The relations presented in this appendix were used to calculate the relevant physical and thermodynamic properties of FC-87. All relations, except that for the vapor dynamic viscosity, were generated from curve fits of property data published by 3M Corporation. The vapor viscosity is unknown, and is therefore estimated using Lucas' method, which is presented below. The data are valid for $20 < T < 80^{\circ}\text{C}$. In the expressions which follow, P is given in bars and T in $^{\circ}\text{C}$.

A.1 Property Relations

1. Saturation Temperature, $T_{\text{sat}}(P)$:

$$T_{\text{sat}} (^{\circ}\text{C}) = -8.58181276 + 49.9516086 P - 15.06236247 P^2 + 2.717311907 P^3 - 0.1962241098 P^4 \quad (\text{A.1})$$

2. Saturation Pressure, $P_{\text{sat}}(T)$:

$$P_{\text{sat}} (\text{bar}) = 0.2495501529 + 0.01885309262 T + 0.0001239063032 T^2 + 4.18978433 \times 10^{-6} T^3 \quad (\text{A.2})$$

3. Density, liquid, ρ_l :

$$\rho_l (\text{kg/m}^3) = 1949.52224 - 10.16562488 T + 0.1692868739 T^2 - 0.002136889443 T^3 + 8.045897135 \times 10^{-6} T^4 \quad (\text{A.3})$$

4. Density, vapor, ρ_g :

$$\rho_g (\text{kg/m}^3) = 2.281459716 + 0.3231243337 T - 0.00126417128 T^2 + 7.361978099 \times 10^{-5} T^3 \quad (\text{A.4})$$

5. Enthalpy, liquid, h_f :

$$h_f \text{ (kJ/kg)} = -0.01769359364 + 1.12231746 T - 0.00230599836 T^2 + 1.846219316 \times 10^{-5} T^3 \quad (\text{A.5})$$

6. Enthalpy of vaporization, h_{fg} :

$$h_{fg} \text{ (kJ/kg)} = 101.3279152 - 0.507085092 T + 0.003299816463 T^2 - 2.258900124 \times 10^{-5} T^3 \quad (\text{A.6})$$

7. Vapor enthalpy, h_g :

$$h_g \text{ (kJ/kg)} = 100.9158578 + 0.6436365089 T + 0.0003768818698 T^2 \quad (\text{A.7})$$

8. Coefficient of Surface Tension, σ :

$$\sigma \times 10^3 \text{ (N/m)} = 12.55031729 - 0.1220063458 T \quad (\text{A.8})$$

9. Liquid conductivity, k_f :

$$k_f \times 10^3 \text{ (W/m K)} = 59.90037017 - 0.1564500264 T \quad (\text{A.9})$$

10. Liquid dynamic viscosity, μ_f :

$$\mu_f \times 10^6 \text{ (kg/m s)} = 697.382533 - 10.65949228 T + 0.09782017889 T^2 - 0.0004165597581 T^3 \quad (\text{A.10})$$

11. Liquid specific heat, $C_{p,f}$:

$$C_{p,f} \text{ (kJ/kg K)} = 1.03617927 + 0.001805922792 T \quad (\text{A.11})$$

12. Vapor dynamic viscosity, μ_g :

$$\mu_g \text{ (kg/m s)} = 1.061691986 \times 10^{-5} + 3.870019716 \times 10^{-8} T \quad (\text{A.12})$$

A.2 Estimation of FC-87 Vapor Viscosity

The vapor dynamic viscosity, μ_g , of FC-87 is not available in the literature and is therefore estimated in the following manner. Following Reid et al. (1987), Lucas' method of corresponding states is used to estimate μ_g ,

$$\xi\mu_g = [0.807T_r^{0.618} - 0.357e^{-0.449T_r} + 0.340e^{-4.058T_r} + 0.018]F_p^*F_Q^* \quad , \quad (A.13)$$

where T_r is the reduced temperature, T/T_c , where T_c is the critical temperature. F_p^* is a correction factor for polar gases, and F_Q^* is a correction factor only for the gases He, H₂ and D₂. ξ is defined by

$$\xi = 0.176 \left(\frac{T_c}{M^3 P_c^4} \right)^{1/6} \quad , \quad (A.14)$$

where ξ is given in μP^{-1} (micropoise⁻¹), T_c in kelvins, M is the molecular weight in kg/kmol, and P_c is the critical pressure given in N/m². According to Nebergall et al. (1980), fluids with high dielectric constants tend to be more polar; for example, water, with a dielectric constant, κ , of 80 is highly polar, whereas carbon tetrachloride ($\kappa=2.2$) is non-polar. By this reasoning it is assumed that FC-87 ($\kappa = 1.73$) is non-polar. Therefore the constants F_p^* and F_Q^* both have values of unity in (B.1).

Evidence of this method's effectiveness can be found by comparing the estimate of μ_g with available data for FC-72, a perfluorocarbon with similar characteristics as FC-87. Comparison of Lucas' technique with a curve fit of the vapor viscosity data for FC-72 yielded percent errors less than 2 percent over the temperature range $-100 < T < 100^\circ C$. The method is therefore judged to be suitable for this fluid. The following relation was then constructed to estimate μ_g over the temperature range $20 < T < 80^\circ C$:

$$\mu_g \text{ (kg/m s)} = 1.061691986 \times 10^{-5} + 3.870019716 \times 10^{-8} T, \quad (\text{A.15})$$

where T is given in °C.

APPENDIX B ESTIMATION OF SHEAR LIFT FORCE ON VAPOR BUBBLE

Following Mei and Klausner (1994), an interpolation formula is developed to obtain an expression for the shear lift force on a vapor bubble applicable over a range of bubble Reynolds number. Legendre and Magnaudet (1997) correctly derived an expression for the lift force on a spherical bubble exposed to a viscous shear flow, in the limit as $Re_b \rightarrow 0$:

$$F_L = \frac{9}{\pi} \mu \Delta U a (Re_b \alpha / 2)^{1/2} J(\sqrt{2\alpha/Re_b}) \quad (B.1)$$

where $\alpha = |(dU/dy)(a/\Delta U)|$ is the dimensionless shear rate. A curve fit of the function $J(\epsilon)$ was obtained by Mei (1992) for $0.1 \leq \epsilon \leq 20$ as

$$J(\epsilon) = 0.6765 \{1 + \tanh[2.5(\log_{10} \epsilon + 0.191)]\} \{0.667 + \tanh[6(\epsilon - 0.32)]\} \quad (B.2)$$

Defining

$$C_L = \frac{F_L}{\frac{1}{2} \pi a^2 \rho (\Delta U)^2}$$

as the lift coefficient, Eq. (A1) is written as

$$C_L = \frac{1.146 J(\sqrt{2\alpha/Re_b})}{Re_b^{1/2}} \quad (B.3)$$

For inviscid flow, Auton (1987) derived an expression for the shear lift coefficient in the limit as $Re_b \rightarrow \infty$:

$$C_L = \frac{3}{4} \alpha \quad (B.4)$$

Using the result of Auton (1987) for the large Reynolds number limit and the result of Legendre and Magnaudet (1997) for the small Reynolds number limit, the following interpolation formula has been obtained for the shear lift coefficient:

$$C_L = \alpha^{1/2} \left\{ \left(\frac{1.146 J(\sqrt{2\alpha}/Re_b)}{Re_b^{1/2}} \right)^n + \left(\frac{3}{4} \alpha^{1/2} \right)^n \right\}^{1/n}, \quad n = 2 \quad (B.5)$$

The above follows the analytical results only in the limits $Re_b \rightarrow 0$ or $Re_b \rightarrow \infty$ with small α ; the result is speculative in the intermediate ranges of Re and α . Consistent with Mei and Klausner (1994), the value $n=2$ is chosen based on the expectation that transition between these two limits occurs at $Re_b \approx 1-10$, since in this range of Re_b the flow is neither viscous dominant nor inviscid. Figure 11 depicts C_L as a function of bubble Reynolds number for various values of α . As observed from the graph, the limiting values of C_L are recovered, and transition indeed takes place between $Re_b \approx 1-10$.

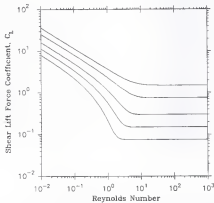


Figure B.1. Shear lift coefficient C_L as calculated from Equation (B.5).

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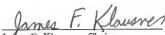
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BIOGRAPHICAL SKETCH

Glen was born on June 30, 1966, in Elgin, Illinois, the third son of Gene and Ruth Thorncroft. From the age of six Glen lived with his family in Orlando, Florida, where he obtained his Bachelor of Science and Master of Science Degrees in Mechanical Engineering at the University of Central Florida in 1988 and 1991, respectively. During his tenure, Glen served as a design engineer at Martin Marietta Corporation, as well as a summer intern at both Florida Power Corporation and NASA Kennedy Space Center. In 1992, Glen began his studies at the University of Florida, in pursuit of his Ph.D. in Mechanical Engineering. He has been supported since 1994 as a Florida Space Grant Fellow.

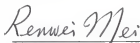
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